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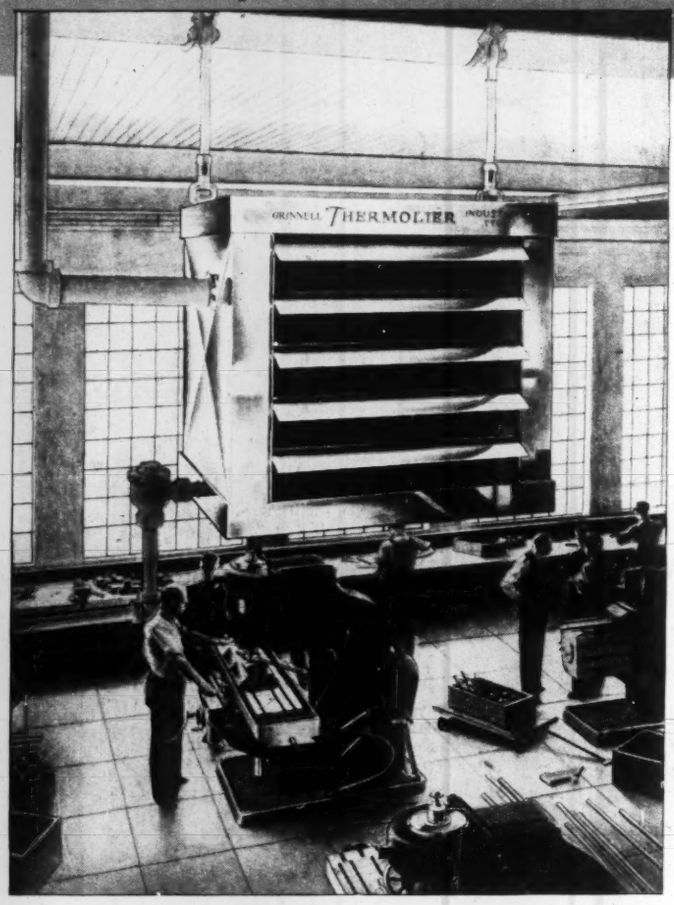

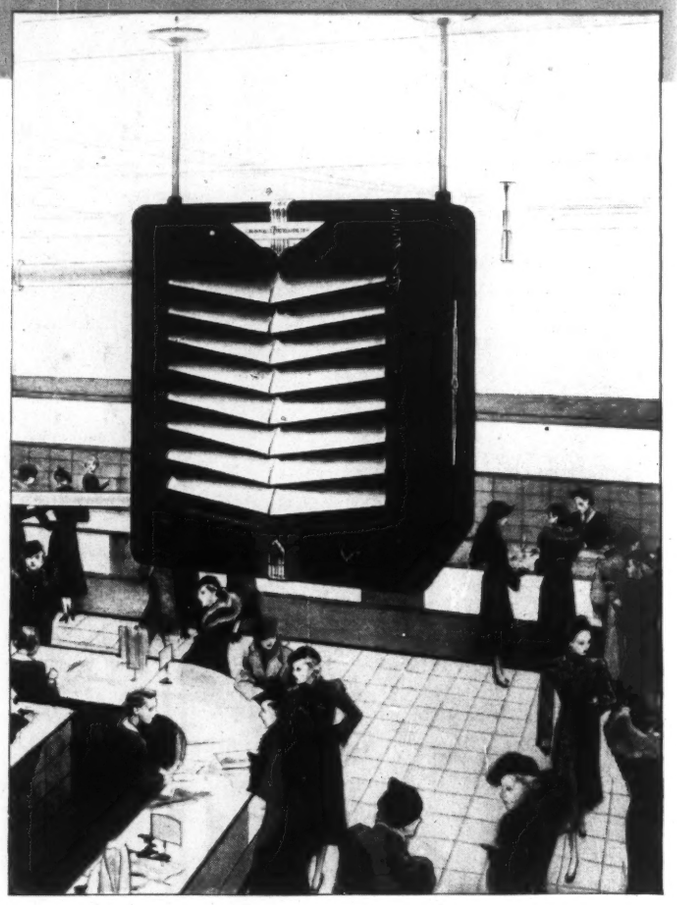

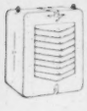

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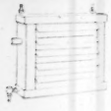

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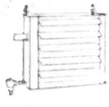

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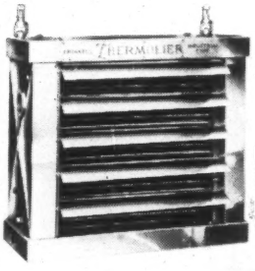

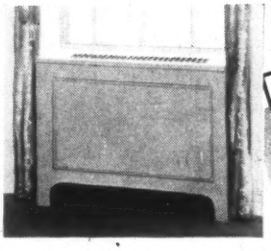

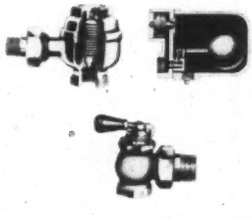

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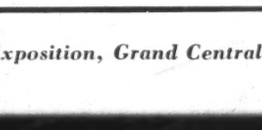

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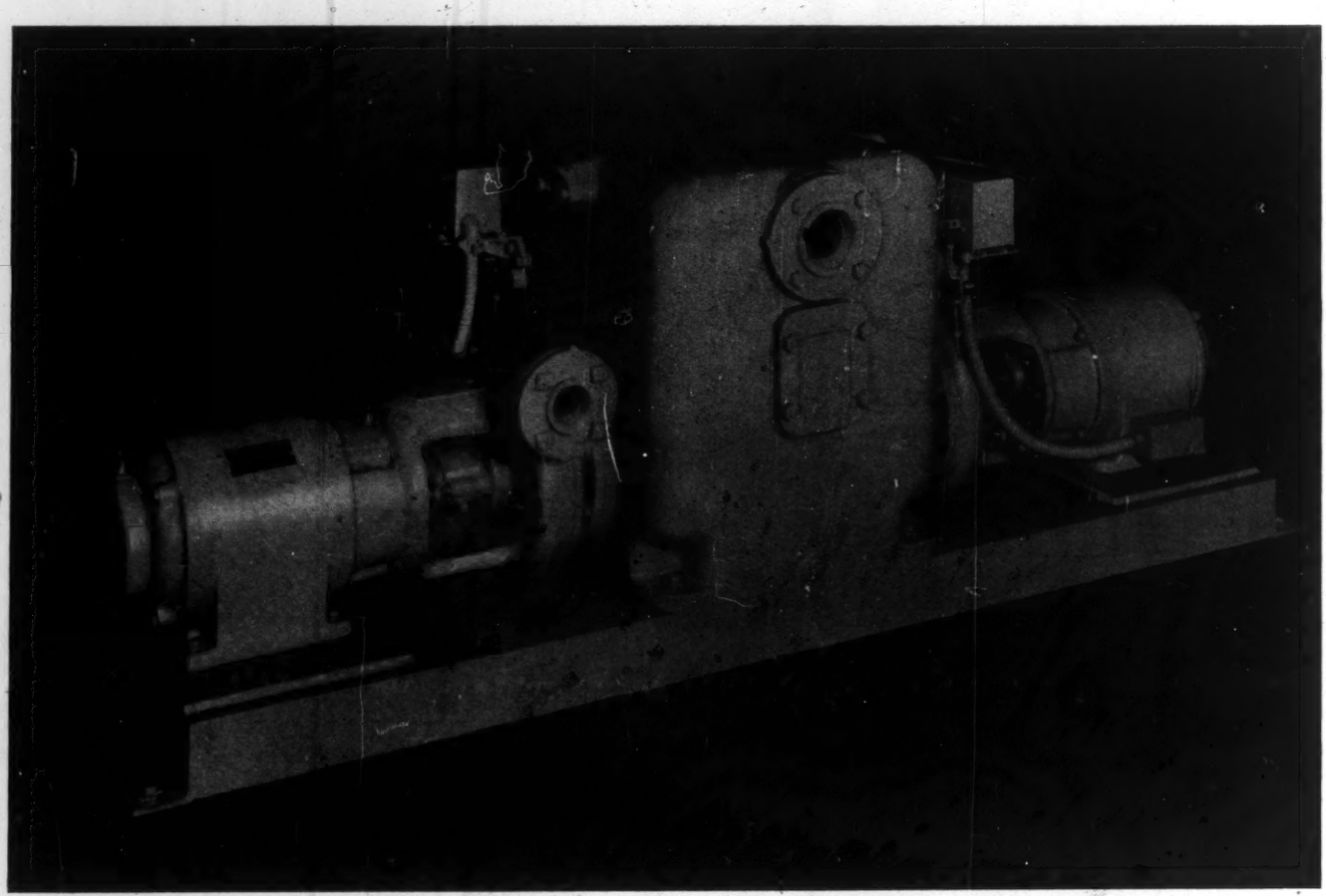
MAXIMUM EFFICIENCY FOR STEAM HEATING SYSTEMS ON HEATING, DRYING AND PROCESS WORK.



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Heating · Piping and Air Conditioning

Vol. 10, *No.* 1 • *January*, 1938

New Tables of the Psychrometric Properties of Air-Vapor Mixtures

By William Goodman*

The editors are privileged to present here new and accurate tables of the psychrometric properties of air-vapor mixtures compiled by Mr. Goodman and based upon the new Keenan-Keyes steam tables . . . In his discussion of their use, the author clarifies the subject of heat balances, which is basic to all air conditioning computations

THE psychrometric tables presented here have been computed from the data in the new Keenan and Keyes steam tables; the text not only describes the use of these tables and how they were computed, but also describes the method of computing the heat to be removed when cooling and dehumidifying air. Although this computation is simple, there seems—in general—to be some confusion in regard to the correct procedure. This is particularly true when the moisture condensed from the air freezes on the cold cooling surface.

Most of this confusion seems to be due to the use of approximate methods of computation. Although approximate methods result in numerical answers that are close enough for many practical purposes, there are occasions when accurate computations are desirable. In this article, the correct method will be presented first, after which the two commonly-used approximate methods will be discussed. The approximate methods are used by many because they shorten the work of computation slightly.

Before proceeding to a discussion of these methods, an explanation is necessary of the term *enthalpy* and the method of computing the enthalpy of air-vapor mixtures.

Enthalpy of Air-Vapor Mixtures

There are excellent reasons for abandoning the phrase *heat content* in favor of the term *enthalpy*[1], and this is occurring in all branches of thermodynamics. For the purposes of this article, in which only constant-pressure processes are discussed, the term enthalpy may be regarded as synonymous with the familiar term heat content.

The enthalpy of dry air, above a base temperature of 0 F, is given by the following familiar equation:

$$h_a = 0.24t \quad \ldots\ldots [1]$$

The enthalpy of the vapor mixed with the air, above a base temperature of 32 F, is given by:

$$h_v = wh_g \quad \ldots\ldots [2]$$

Adding [1] and [2], the equation for the enthalpy of an air-vapor mixture is obtained:

$$h = h_a + h_v$$

$$h = 0.24t + wh_g \quad \ldots\ldots [3]$$

Equation 3 is generally applicable regardless of whether the air is fully or partially saturated. If the air is partially saturated, the dry bulb temperature is used in the first term, and the enthalpy of the vapor corresponding to the same dry bulb temperature is used in the second term. However, the value of w in the second term corresponds to the actual dew point temperature. On the other hand, if the air is "100 per cent saturated", all of the quantities correspond to the same temperature, which is of course the saturation temperature of the mixture.

When using [3], values of w corresponding to the given dew point temperatures are obtained from column 4 of the Psychrometric Tables; values of h_g, the enthalpy per pound of saturated vapor corresponding to the dry bulb temperature t, are obtained

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[1]The nomenclature of psychrometry is not altogether satisfactory. Such terms as the "moisture content of air," "saturated air," etc., are misleading insofar as basic physical ideas are concerned. Although some attempt has been made to use more rational terms, it is difficult to do so at times without sacrificing the clarity that is obtained by the use of familiar terms. For instance, the phrase "wet bulb temperature" has been used instead of the correct "temperature of adiabatic saturation," because the latter is not widely used. Incidentally, the two are not identical, but for air-vapor mixtures the difference in their numerical values is very small.

from the Keenan and Keyes steam tables, or from equation 4 which is given later.

Values of h_a, computed by means of equation 1, are tabulated in column 9, and values of h_v, computed by means of [2], are tabulated in column 10. Values of h, tabulated in column 11, were obtained in accordance with [3] by adding the values in columns 9 and 10. It should be emphasized that column 11 gives the enthalpy of *saturated* air-vapor mixtures only; the enthalpy of unsaturated mixtures is discussed later.

Enthalpy of Water Vapor

At low pressure, water vapor may be considered as a perfect gas, in which case its enthalpy depends only upon its temperature. For pressures below 5 lb absolute and temperatures below 500 F, the following empirical formula gives values of the enthalpy of water vapor which deviate by less than ¼ of 1 per cent from those given by the Keenan and Keyes steam tables:

$$h_g = 1061 + 0.45t \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [4]$$

Even for 200 F saturated steam having a pressure of 11.53 lb absolute, the deviation from the value given by the Keenan and Keyes steam tables is less than ½ of 1 per cent.

Above 32 F, the actual values of enthalpy given by the Keenan and Keyes tables were used in equation 2 for computing the values in column 10. Below 32 F, the values of enthalpy were computed by means of equation 4.

Symbols for Psychrometric Tables and Text

d	Density of vapor	Lb per cu ft
h	Enthalpy of air-vapor mixture	Btu per lb of dry air
h'	Enthalpy of a *saturated* mixture at the wet bulb temperature	Btu per lb of dry air
h''	Enthalpy of a *saturated* mixture at the dew point temperature	Btu per lb of dry air
h_a	Enthalpy of air only, in an air-vapor mixture	Btu per lb of dry air
h_t	Enthalpy of water or ice at the dry bulb temperature	Btu per lb of steam
h'_t	Enthalpy of water or ice at the wet bulb temperature	Btu per lb of water
h''_t	Enthalpy of water or ice at the dew point temperature	Btu per lb of water
h_g	Enthalpy of vapor at the dry bulb temperature	Btu per lb of steam
h'_g	Enthalpy of vapor at the wet bulb temperature	Btu per lb of steam
h''_g	Enthalpy of vapor at the dew point temperature	Btu per lb of steam
h_o	Enthalpy of make-up water supplied to, or of condensate or ice removed from, air conditioning apparatus	Btu per lb of water
h_v	Enthalpy of vapor only, in an air-vapor mixture	Btu per lb of dry air
p_v	Vapor pressure of water	Lb per sq in.
P	Total barometric pressure	Lb per sq in.
q	Heat removed from air-vapor mixture by external means, such as chilled water or direct expansion refrigerant	Btu per lb of dry air
r'	Enthalpy of either vaporization or sublimation	Btu per lb of steam
s	Humid specific heat of air-vapor mixture	Btu per lb of dry air per deg Fahrenheit
s'	Humid specific heat of a *saturated* mixture at the wet bulb temperature	Btu per lb of dry air per deg Fahrenheit
s''	Humid specific heat of a *saturated* mixture at the dew point temperature	Btu per lb of dry air per deg Fahrenheit
t	Dry bulb temperature of air-vapor mixture	Deg Fahrenheit
t'	Wet bulb temperature of air-vapor mixture	Deg Fahrenheit
t''	Dew point temperature of air-vapor mixture	Deg Fahrenheit
T	Absolute temperature	Deg Fahrenheit
v	Specific volume of air-vapor mixture	Cu ft per lb of dry air
v_D	Specific volume of dry air	Cu ft per lb of dry air
w	Absolute humidity (weight of moisture in an air-vapor mixture. Also called the moisture content)	Lb per lb of dry air
w'	Absolute humidity at the wet bulb temperature	Lb per lb of dry air

Numerical Subscripts—Numerical subscripts (such as 1 and 2) are used to indicate the point in the air conditioning cycle at which the quantity is to be taken. For example, t_1 indicates the initial dry bulb temperature of the air; t'_1, the initial wet bulb; t_2 the final dry bulb; t'_2, the final wet bulb; h_1, the initial enthalpy of the air; w_2 the final absolute humidity; and h'_2, the enthalpy of saturated air at the final wet bulb temperature.

Enthalpy of Ice and Low Temperature Water Vapor

Temp. Deg Fahr.	Enthalpy Btu per Lb of Ice or Vapor		
	Ice	Sublimation	Saturated Vapor
1	2	3	4
—40	—177.55	1220.6	1043.0
—39	—177.08	1220.6	1043.5
—38	—176.60	1220.6	1043.9
—37	—176.13	1220.5	1044.4
—36	—175.65	1220.5	1044.8
—35	—175.18	1220.5	1045.3
—34	—174.70	1220.5	1045.7
—33	—174.23	1220.4	1046.2
—32	—173.75	1220.4	1046.6
—31	—173.28	1220.4	1047.1
—30	—172.80	1220.4	1047.5
—29	—172.33	1220.3	1048.0
—28	—171.85	1220.3	1048.4
—27	—171.38	1220.3	1048.9
—26	—170.90	1220.3	1049.3
—25	—170.43	1220.2	1049.8
—24	—169.95	1220.2	1050.2
—23	—169.48	1220.2	1050.7
—22	—169.00	1220.2	1051.1
—21	—168.53	1220.1	1051.6
—20	—168.05	1220.1	1052.0
—19	—167.58	1220.1	1052.5
—18	—167.10	1220.1	1052.9
—17	—166.63	1220.0	1053.4
—16	—166.15	1220.0	1053.8
—15	—165.68	1220.0	1054.3
—14	—165.20	1220.0	1054.7
—13	—164.73	1219.9	1055.2
—12	—164.25	1219.9	1055.6
—11	—163.78	1219.9	1056.1
—10	—163.30	1219.9	1056.5
— 9	—162.83	1219.8	1057.0
— 8	—162.35	1219.8	1057.4
— 7	—161.88	1219.8	1057.9
— 6	—161.40	1219.8	1058.3
— 5	—160.93	1219.7	1058.8
— 4	—160.45	1219.7	1059.2
— 3	—159.98	1219.7	1059.7
— 2	—159.50	1219.7	1060.1
— 1	—159.03	1219.6	1060.6
0	—158.55	1219.6	1061.0
1	—158.06	1219.6	1061.5
2	—157.60	1219.6	1061.9
3	—157.13	1219.5	1062.4
4	—156.65	1219.5	1062.8
5	—156.18	1219.5	1063.3
6	—155.70	1219.5	1063.7
7	—155.23	1219.4	1064.2
8	—154.75	1219.4	1064.6
9	—154.28	1219.4	1065.1
10	—153.80	1219.4	1065.5
11	—153.33	1219.3	1066.0
12	—152.85	1219.3	1066.4
13	—152.38	1219.3	1066.9
14	—151.90	1219.3	1067.3
15	—151.43	1219.2	1067.8
16	—150.95	1219.2	1068.2
17	—150.48	1219.2	1068.7
18	—150.00	1219.2	1069.1
19	—149.53	1219.1	1069.6
20	—149.05	1219.1	1070.0
21	—148.58	1219.1	1070.5
22	—148.10	1219.1	1070.9
23	—147.63	1219.0	1071.4
24	—147.15	1219.0	1071.8
25	—146.68	1219.0	1072.3
26	—146.20	1219.0	1072.7
27	—145.73	1218.9	1073.2
28	—145.25	1218.9	1073.6
29	—144.78	1218.9	1074.1
30	—144.30	1218.9	1074.5
31	—143.83	1218.8	1075.0
32	—143.35	1218.8	1075.4

The tabulated values in this table were computed by means of the following formulas:

Enthalpy of Ice (in column 2):

$$h_t = -158.55 + 0.475t$$

Within the range of the table, this formula agrees with the Keenan and Keyes data with a maximum deviation of ± 0.3 of 1 per cent

Enthalpy of Sublimation (in column 3):

$$r = 1219.6 - 0.025t$$

Within the range of the table, this formula agrees with the Keenan and Keyes data with a maximum deviation of —0.1 of 1 per cent.

The enthalpy of the saturated vapor, tabulated in column 4, was computed by equation [4] of the text.

Correct Method of Computing Heat Removed

Let the rectangle in Fig. 1 represent any kind of a perfectly insulated air conditioning apparatus for cooling and dehumidifying air. The heat carried out of the apparatus by the streams of air and condensate are shown by the arrows. Evidently, for a heat balance, the total heat entering and leaving the apparatus must be equal, or basing the analysis upon 1 lb of air,

$$h_1 = h_2 + (w_1 - w_2)\, h_o + q$$
$$q = (h_1 - h_2) - (w_1 - w_2)\, h_o \quad \ldots\ldots\ldots\ldots\ldots\ldots [5]$$

Equation 5 shows that the total heat which must be removed in cooling and dehumidifying air is *less* than the difference between the initial and final enthalpies of the air. This statement is true as long as the temperature of the condensate draining from the apparatus is above 32 F. If freezing of the moisture condensed from the air occurs, the term h_o in equation 5 will be negative, as shown in column 2 of the table for Enthalpy of Ice and Low Temperature Water Vapor. Hence, for this last condition, the total heat removed will be *greater* than the difference between the initial and final enthalpies of the air. *A correct answer for the total heat to be removed is always obtained by using equation 5, whether or not the moisture condensed from the air freezes.*

To some, the inconvenience of using [5] arises from the fact that two of the three temperatures—dry bulb, wet bulb, or dew point—must be known before the enthalpy of the air-vapor mixture can be computed. The enthalpy cannot be found if the wet bulb temperature alone is known. This is apparently in contradiction to the well-known fact that "the total heat depends only upon the wet bulb temperature." The answer to this is that the quantity commonly used as the total heat is *not*, in reality, the total heat or enthalpy of the mixture; it is a quantity that differs slightly in numerical value from the true enthalpy. In order to make a clear distinction between this commonly used quantity and the true enthalpy, Carrier and Mackey[2] have recently renamed it the "sigma function" because of the Greek letter, Σ, which they use to represent it. Now, a true statement is "The sigma function depends only upon the wet bulb temperature of the air". The sigma function will be discussed later in connection with the two approximate computation methods previously mentioned. This digression was made only to anticipate any possible confusion which might arise because of the fact stated previously, that the enthalpy of an air-vapor mixture cannot be found if the wet bulb temperature alone is known.

The psychrometric tables list only the enthalpy of saturated air (column 11). However, the enthalpy of unsaturated air can be computed by means of the tables and any one of the three formulas below.

Dry bulb and dew point temperatures given:

$$h = h'' + s''\,(t - t'') \quad \ldots\ldots\ldots\ldots\ldots\ldots [6]$$

Wet bulb and dew point temperatures given:

$$h = h' - h'_f\,(w' - w) \quad \ldots\ldots\ldots\ldots\ldots\ldots [7]$$

Dry bulb and wet bulb temperatures given:

$$h = h' - \frac{s'\,(t - t')}{\frac{h_g}{h'_f} - 1} \quad \ldots\ldots\ldots\ldots\ldots\ldots [8]$$

Values of s'' or s' are tabulated in column 8, and values of h'' or h' in column 11. Values of h_g can be found in the Keenan and Keyes steam tables, or evaluated by equation 4. Values of h'_f are found by means of the relation, $h'_f = t' - 32$.

If the dry bulb and dew point temperatures are given, there are also two other equations in addition to [6] that can be used for finding the enthalpy.

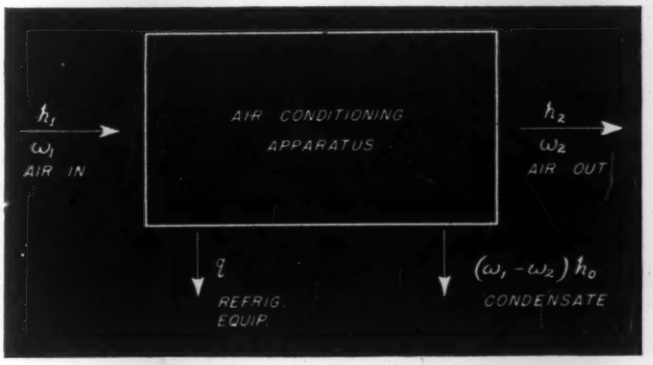


Fig. 1—Diagram representing a perfectly insulated air conditioning apparatus, showing heat balance

From [3] and [1],

$$h = h_a + wh_g \quad \ldots\ldots\ldots\ldots\ldots\ldots [3a]$$

From [3] and [4],

$$h = s''t + 1061w \quad \ldots\ldots\ldots\ldots\ldots\ldots [3b]$$

If an accurate psychrometric chart is used only one of the preceding equations need be memorized, as the temperatures required for any one of the preceding equations can then be found no matter which temperatures are given originally.

The two examples which follow illustrate the use of equations 5 and 7. In the first example, the final dew point temperature of the air is above 32 F. In the second example, the dew point is below 32 F and it is assumed that the condensate freezes. After following through these two examples, the use of equations 6 and 8 will also be obvious.

Example 1: Humid air having an initial wet bulb temperature of 75 F and a dew point temperature of 60 F is cooled to a final wet bulb of 55 F and a dew point of 50 F. The temperature of the condensed moisture leaving through the drain is 52 F as measured by a thermometer. Find the total heat removed by the refrigerant.

Solution: The initial and final enthalpies must first be computed by equation 7.

From col. 11, for 75 deg W.B., $h'_1 = 38.53$ Btu.
From $(t' - 32)$, for 75 deg W.B., $h'_{f1} = 43.00$ Btu.
From col. 4, for 75 deg W.B., $w'_1 = 0.01876$ lb.
From col. 4, for 60 deg D.P., $w_1 = 0.01105$ lb.
By equation 7, $h_1 = 38.53 - 43\,(0.01876 - 0.01105)$
$= 38.198$ Btu.
Similarly, by equation 7, $h_2 = 23.19 - 23\,(0.00920 - 0.007637)$
$= 23.154$ Btu.

As the initial and final enthalpies of the air-vapor mixture are now known, it is possible to use equation 5 to compute the total heat removed.

From column 4, for a final D.P. temperature of 50 deg, $w_2 = 0.007637$ lb.

[Continued on p. 6]

[2] "A Review of Existing Psychrometric Data in Relation to Practical Engineering Problems," by W. H. Carrier and C. O. Mackey, *Transactions*, American Society of Mechanical Engineers, Vol. 59, 1937, p. 33.

Properties of Mixtures of Air and Saturated Water Vapor

Barometric Pressure 14.696 Lb per Sq In.

All tabulated values (except temperatures and pressures) are quantities per pound of dry air in the mixture.

Temp. Deg Fahr. t	Pressure: Lb per Sq In. p_v	Pressure: Inches of Mercury p_v	Absolute Humidity: Pounds w	Absolute Humidity: Grains w	Volume: Dry Air Cu Ft v_D	Volume: Increase for Humid Air *Per Cent*	Humid Specific Heat s	Enthalpy (Heat Content): Dry Air Btu h_a	Enthalpy (Heat Content): Water Vapor Btu h_v	Enthalpy (Heat Content): Air-Vapor Mixture Btu h	Temp. Deg Fahr. t
1	2	3	4	5	6	7	8	9	10	11	12
— 40	0.002749	0.005587	0.0001164	0.8146	10.58	0.019	0.2401	— 9.60	0.1214	— 9.479	— 40
— 39	0.002910	0.005925	0.0001232	0.8623	10.60	0.020	0.2401	— 9.36	0.1285	— 9.232	— 39
— 38	0.003083	0.006277	0.0001305	0.9136	10.63	0.021	0.2401	— 9.12	0.1362	— 8.984	— 38
— 37	0.003264	0.006646	0.0001382	0.9673	10.65	0.022	0.2401	— 8.88	0.1443	— 8.736	— 37
— 36	0.003455	0.007034	0.0001463	1.024	10.68	0.024	0.2401	— 8.64	0.1528	— 8.487	— 36
— 35	0.003653	0.007438	0.0001547	1.083	10.71	0.025	0.2401	— 8.40	0.1617	— 8.238	— 35
— 34	0.003866	0.007871	0.0001637	1.146	10.73	0.026	0.2401	— 8.16	0.1712	— 7.989	— 34
— 33	0.004087	0.008321	0.0001730	1.211	10.76	0.028	0.2401	— 7.92	0.1810	— 7.739	— 33
— 32	0.004322	0.008800	0.0001830	1.281	10.78	0.029	0.2401	— 7.68	0.1915	— 7.489	— 32
— 31	0.004564	0.009292	0.0001932	1.353	10.81	0.031	0.2401	— 7.44	0.2023	— 7.238	— 31
— 30	0.004825	0.009824	0.0002043	1.430	10.83	0.033	0.2401	— 7.20	0.2140	— 6.986	— 30
— 29	0.005096	0.01038	0.0002158	1.510	10.86	0.035	0.2401	— 6.96	0.2261	— 6.734	— 29
— 28	0.005376	0.01095	0.0002276	1.593	10.88	0.037	0.2401	— 6.72	0.2386	— 6.481	— 28
— 27	0.005678	0.01156	0.0002404	1.683	10.91	0.039	0.2401	— 6.48	0.2522	— 6.228	— 27
— 26	0.005989	0.01219	0.0002536	1.775	10.93	0.041	0.2401	— 6.24	0.2661	— 5.974	— 26
— 25	0.006317	0.01286	0.0002675	1.872	10.96	0.043	0.2401	— 6.00	0.2808	— 5.719	— 25
— 24	0.006663	0.01357	0.0002821	1.975	10.98	0.045	0.2401	— 5.76	0.2963	— 5.464	— 24
— 23	0.007028	0.01431	0.0002976	2.083	11.01	0.048	0.2401	— 5.52	0.3127	— 5.207	— 23
— 22	0.007405	0.01508	0.0003136	2.195	11.03	0.050	0.2401	— 5.28	0.3296	— 4.950	— 22
— 21	0.007802	0.01588	0.0003304	2.313	11.06	0.053	0.2402	— 5.04	0.3474	— 4.693	— 21
— 20	0.008221	0.01674	0.0003481	2.437	11.08	0.056	0.2402	— 4.80	0.3662	— 4.434	— 20
— 19	0.008661	0.01763	0.0003668	2.568	11.11	0.059	0.2402	— 4.56	0.3860	— 4.174	— 19
— 18	0.009115	0.01856	0.0003860	2.702	11.13	0.062	0.2402	— 4.32	0.4065	— 3.914	— 18
— 17	0.009593	0.01953	0.0004063	2.844	11.16	0.065	0.2402	— 4.08	0.4280	— 3.652	— 17
— 16	0.01010	0.02056	0.0004278	2.994	11.18	0.069	0.2402	— 3.84	0.4508	— 3.389	— 16
— 15	0.01062	0.02162	0.0004498	3.149	11.21	0.072	0.2402	— 3.60	0.4742	— 3.126	— 15
— 14	0.01117	0.02274	0.0004731	3.312	11.23	0.076	0.2402	— 3.36	0.4990	— 2.861	— 14
— 13	0.01175	0.02392	0.0004977	3.484	11.26	0.080	0.2402	— 3.12	0.5252	— 2.595	— 13
— 12	0.01235	0.02515	0.0005231	3.662	11.28	0.084	0.2402	— 2.88	0.5522	— 2.328	— 12
— 11	0.01297	0.02641	0.0005494	3.846	11.31	0.088	0.2403	— 2.64	0.5802	— 2.060	— 11
— 10	0.01364	0.02777	0.0005778	4.045	11.34	0.093	0.2403	— 2.40	0.6105	— 1.790	— 10
— 9	0.01432	0.02916	0.0006067	4.247	11.36	0.098	0.2403	— 2.16	0.6412	— 1.519	— 9
— 8	0.01505	0.03064	0.0006376	4.463	11.39	0.103	0.2403	— 1.92	0.6742	— 1.246	— 8
— 7	0.01579	0.03215	0.0006690	4.683	11.41	0.108	0.2403	— 1.68	0.7077	— 0.9723	— 7
— 6	0.01657	0.03374	0.0007021	4.915	11.44	0.113	0.2403	— 1.44	0.7430	— 0.6970	— 6
— 5	0.01740	0.03543	0.0007373	5.161	11.46	0.119	0.2403	— 1.20	0.7807	— 0.4194	— 5
— 4	0.01825	0.03716	0.0007734	5.414	11.49	0.124	0.2404	— 0.96	0.8192	— 0.1408	— 4
— 3	0.01914	0.03897	0.0008111	5.678	11.51	0.130	0.2404	— 0.72	0.8595	+ 0.1394	— 3
— 2	0.02008	0.04088	0.0008510	5.957	11.54	0.137	0.2404	— 0.48	0.9022	0.4222	— 2
— 1	0.02106	0.04288	0.0008926	6.248	11.56	0.144	0.2404	— 0.24	0.9467	0.7067	— 1
0	0.02206	0.04491	0.0009351	6.546	11.59	0.150	0.2404	0	0.9921	0.9921	0
1	0.02311	0.04705	0.0009797	6.858	11.61	0.158	0.2404	0.24	1.040	1.280	1
2	0.02421	0.04929	0.001026	7.185	11.64	0.165	0.2405	0.48	1.090	1.570	2
3	0.02536	0.05163	0.001075	7.526	11.66	0.173	0.2405	0.72	1.142	1.862	3
4	0.02657	0.05410	0.001127	7.886	11.69	0.181	0.2405	0.96	1.197	2.157	4
5	0.02781	0.05662	0.001179	8.255	11.71	0.190	0.2405	1.20	1.254	2.454	5
6	0.02910	0.05925	0.001234	8.639	11.74	0.198	0.2406	1.44	1.313	2.753	6
7	0.03045	0.06200	0.001292	9.041	11.76	0.208	0.2406	1.68	1.374	3.054	7
8	0.03186	0.06487	0.001351	9.460	11.79	0.217	0.2406	1.92	1.439	3.359	8
9	0.03334	0.06788	0.001414	9.900	11.81	0.227	0.2406	2.16	1.506	3.666	9
10	0.03485	0.07096	0.001479	10.35	11.84	0.238	0.2407	2.40	1.575	3.975	10
11	0.03647	0.07425	0.001547	10.83	11.86	0.249	0.2407	2.64	1.650	4.290	11
12	0.03812	0.07761	0.001618	11.32	11.89	0.260	0.2407	2.88	1.725	4.605	12
13	0.03980	0.08103	0.001689	11.82	11.92	0.272	0.2408	3.12	1.802	4.922	13
14	0.04163	0.08476	0.001767	12.37	11.94	0.284	0.2408	3.36	1.886	5.246	14
15	0.04349	0.08855	0.001846	12.92	11.97	0.297	0.2408	3.60	1.971	5.571	15
16	0.04541	0.09246	0.001925	13.47	11.99	0.310	0.2409	3.84	2.056	5.896	16
17	0.04741	0.09653	0.002013	14.09	12.02	0.324	0.2409	4.08	2.151	6.231	17
18	0.04950	0.1008	0.002102	14.72	12.04	0.338	0.2410	4.32	2.247	6.567	18
19	0.05168	0.1052	0.002195	15.37	12.07	0.353	0.2410	4.56	2.348	6.908	19
20	0.05391	0.1098	0.002290	16.03	12.09	0.368	0.2410	4.80	2.450	7.250	20
21	0.05628	0.1146	0.002391	16.74	12.12	0.384	0.2411	5.04	2.560	7.600	21
22	0.05869	0.1195	0.002494	17.46	12.14	0.401	0.2411	5.28	2.671	7.951	22
23	0.06122	0.1246	0.002602	18.21	12.17	0.418	0.2412	5.52	2.788	8.308	23
24	0.06384	0.1300	0.002714	19.00	12.19	0.436	0.2412	5.76	2.909	8.669	24
25	0.06652	0.1354	0.002828	19.80	12.22	0.455	0.2413	6.00	3.033	9.033	25
26	0.06938	0.1413	0.002950	20.65	12.24	0.474	0.2413	6.24	3.165	9.405	26
27	0.07228	0.1472	0.003074	21.52	12.27	0.494	0.2414	6.48	3.299	9.779	27
28	0.07530	0.1533	0.003203	22.42	12.29	0.515	0.2414	6.72	3.439	10.16	28
29	0.07845	0.1597	0.003338	23.37	12.32	0.537	0.2415	6.96	3.585	10.55	29
30	0.08172	0.1664	0.003478	24.35	12.34	0.559	0.2416	7.20	3.737	10.94	30
31	0.08514	0.1734	0.003625	25.37	12.37	0.583	0.2416	7.44	3.896	11.34	31
32	0.08854	0.1803	0.003771	26.40	12.39	0.606	0.2417	7.68	4.054	11.73	32
33	0.09223	0.1878	0.003930	27.51	12.42	0.632	0.2418	7.92	4.229	12.15	33
34	0.09603	0.1955	0.004092	28.64	12.44	0.658	0.2418	8.16	4.406	12.57	34
35	0.09995	0.2035	0.004260	29.82	12.47	0.685	0.2419	8.40	4.588	12.99	35
36	0.10401	0.2118	0.004435	31.05	12.49	0.713	0.2420	8.64	4.780	13.42	36
37	0.10821	0.2203	0.004616	32.31	12.52	0.742	0.2421	8.88	4.976	13.86	37
38	0.11256	0.2292	0.004803	33.62	12.54	0.772	0.2422	9.12	5.180	14.30	38
39	0.11705	0.2383	0.004996	34.98	12.57	0.803	0.2423	9.36	5.391	14.75	39
40	0.12170	0.2478	0.005196	36.38	12.60	0.835	0.2423	9.60	5.609	15.21	40
41	0.12652	0.2576	0.005403	37.82	12.62	0.867	0.2424	9.84	5.834	15.67	41
42	0.13150	0.2677	0.005619	39.33	12.65	0.903	0.2425	10.08	6.069	16.15	42
43	0.13665	0.2782	0.005840	40.88	12.67	0.939	0.2426	10.32	6.311	16.63	43
44	0.14199	0.2891	0.006070	42.49	12.70	0.976	0.2427	10.56	6.562	17.12	44

Properties of Mixtures of Air and Saturated Water Vapor (Continued)

Barometric Pressure 14.696 Lb per Sq In.

All tabulated values (except temperatures and pressures) are quantities per pound of dry air in the mixture.

	Pressure		Absolute Humidity		Volume			Enthalpy (Heat Content)			
Temp. Deg Fahr. t	Lb per Sq In. p_v	Inches of Mercury p_v	Pounds w	Grains w	Dry Air Cu Ft v_D	Volume Increase for Humid Air *Per Cent*	Humid Specific Heat s	Dry Air Btu h_a	Water Vapor Btu h_v	Air-Vapor Mixture Btu h	Temp. Deg Fahr. t
1	2	3	4	5	6	7	8	9	10	11	12
45	0.14752	0.3004	0.006311	44.18	12.72	1.01	0.2428	10.80	6.826	17.63	45
46	0.15323	0.3120	0.006559	45.91	12.75	1.05	0.2430	11.04	7.096	18.14	46
47	0.15914	0.3240	0.006813	47.69	12.77	1.09	0.2431	11.28	7.375	18.66	47
48	0.16525	0.3364	0.007076	49.53	12.80	1.14	0.2432	11.52	7.662	19.18	48
49	0.17157	0.3493	0.007351	51.45	12.82	1.18	0.2433	11.76	7.962	19.72	49
50	0.17811	0.3626	0.007637	53.46	12.85	1.23	0.2434	12.00	8.276	20.28	50
51	0.18486	0.3764	0.007929	55.51	12.87	1.27	0.2436	12.24	8.596	20.84	51
52	0.19182	0.3906	0.008229	57.60	12.90	1.32	0.2437	12.48	8.925	21.41	52
53	0.19900	0.4052	0.008545	59.82	12.92	1.37	0.2439	12.72	9.272	21.99	53
54	0.20642	0.4203	0.008867	62.07	12.95	1.43	0.2440	12.96	9.624	22.58	54
55	0.2141	0.4359	0.009200	64.40	12.97	1.48	0.2441	13.20	9.989	23.19	55
56	0.2220	0.4520	0.009549	66.84	13.00	1.53	0.2443	13.44	10.37	23.81	56
57	0.2302	0.4686	0.009903	69.32	13.02	1.59	0.2445	13.68	10.76	24.44	57
58	0.2386	0.4858	0.01027	71.92	13.05	1.65	0.2446	13.92	11.17	25.09	58
59	0.2473	0.5035	0.01066	74.59	13.07	1.71	0.2448	14.16	11.59	25.75	59
60	0.2563	0.5218	0.01105	77.32	13.10	1.78	0.2450	14.40	12.02	26.42	60
61	0.2655	0.5407	0.01145	80.17	13.12	1.84	0.2452	14.64	12.47	27.11	61
62	0.2751	0.5601	0.01188	83.16	13.15	1.91	0.2454	14.88	12.94	27.82	62
63	0.2850	0.5802	0.01231	86.20	13.18	1.98	0.2455	15.12	13.41	28.53	63
64	0.2951	0.6009	0.01276	89.30	13.20	2.05	0.2457	15.36	13.90	29.26	64
65	0.3056	0.6222	0.01323	92.60	13.23	2.12	0.2460	15.60	14.42	30.02	65
66	0.3164	0.6442	0.01370	95.90	13.25	2.20	0.2462	15.84	14.94	30.78	66
67	0.3276	0.6669	0.01419	99.36	13.28	2.28	0.2464	16.08	15.49	31.57	67
68	0.3390	0.6903	0.01470	102.9	13.30	2.36	0.2466	16.32	16.05	32.37	68
69	0.3509	0.7144	0.01523	106.6	13.33	2.45	0.2469	16.56	16.63	33.19	69
70	0.3631	0.7392	0.01577	110.4	13.35	2.53	0.2471	16.80	17.23	34.03	70
71	0.3756	0.7648	0.01633	114.3	13.38	2.62	0.2474	17.04	17.85	34.89	71
72	0.3886	0.7912	0.01691	118.4	13.40	2.72	0.2476	17.28	18.49	35.77	72
73	0.4019	0.8183	0.01751	122.6	13.43	2.81	0.2479	17.52	19.15	36.67	73
74	0.4156	0.8462	0.01813	126.9	13.45	2.91	0.2482	17.76	19.83	37.59	74
75	0.4298	0.8750	0.01876	131.3	13.48	3.01	0.2484	18.00	20.53	38.53	75
76	0.4443	0.9046	0.01942	135.9	13.50	3.12	0.2487	18.24	21.26	39.50	76
77	0.4593	0.9352	0.02010	140.7	13.53	3.23	0.2490	18.48	22.01	40.49	77
78	0.4747	0.9666	0.02079	145.5	13.55	3.34	0.2494	18.72	22.78	41.50	78
79	0.4906	0.9989	0.02151	150.6	13.58	3.45	0.2497	18.96	23.58	42.54	79
80	0.5069	1.0321	0.02225	155.8	13.60	3.57	0.2500	19.20	24.40	43.60	80
81	0.5237	1.0664	0.02302	161.1	13.63	3.70	0.2504	19.44	25.26	44.70	81
82	0.5410	1.1016	0.02381	166.6	13.65	3.82	0.2507	19.68	26.14	45.82	82
83	0.5588	1.1378	0.02463	172.4	13.68	3.95	0.2511	19.92	27.04	46.96	83
84	0.5771	1.1750	0.02546	178.2	13.70	4.09	0.2515	20.16	27.97	48.13	84
85	0.5959	1.2133	0.02633	184.3	13.73	4.23	0.2519	20.40	28.93	49.33	85
86	0.6152	1.2527	0.02723	190.6	13.75	4.37	0.2523	20.64	29.93	50.57	86
87	0.6351	1.2931	0.02815	197.0	13.78	4.52	0.2527	20.88	30.95	51.83	87
88	0.6556	1.3347	0.02909	203.6	13.81	4.67	0.2531	21.12	32.00	53.12	88
89	0.6766	1.3775	0.03007	210.5	13.83	4.83	0.2535	21.36	33.10	54.46	89
90	0.6982	1.4215	0.03108	217.6	13.86	4.99	0.2540	21.60	34.22	55.82	90
91	0.7204	1.4667	0.03212	224.9	13.88	5.16	0.2545	21.84	35.38	57.22	91
92	0.7432	1.5131	0.03319	232.3	13.91	5.33	0.2549	22.08	36.57	58.65	92
93	0.7666	1.5608	0.03430	240.1	13.93	5.50	0.2554	22.32	37.81	60.13	93
94	0.7906	1.6097	0.03544	248.1	13.96	5.69	0.2560	22.56	39.08	61.64	94
95	0.8153	1.6600	0.03661	256.3	13.98	5.87	0.2565	22.80	40.39	63.19	95
96	0.8407	1.7117	0.03782	264.8	14.01	6.07	0.2570	23.04	41.74	64.78	96
97	0.8668	1.7647	0.03906	273.4	14.03	6.27	0.2576	23.28	43.12	66.40	97
98	0.8935	1.8192	0.04035	282.5	14.06	6.47	0.2582	23.52	44.57	68.09	98
99	0.9210	1.8751	0.04168	291.7	14.08	6.69	0.2588	23.76	46.04	69.80	99
100	0.9492	1.9325	0.04304	301.3	14.11	6.91	0.2594	24.00	47.57	71.57	100
101	0.9781	1.9915	0.04445	311.2	14.13	7.13	0.2600	24.24	49.15	73.39	101
102	1.0078	2.0519	0.04591	321.4	14.16	7.36	0.2607	24.48	50.78	75.26	102
103	1.0382	2.1138	0.04741	331.9	14.18	7.60	0.2613	24.72	52.46	77.18	103
104	1.0695	2.1775	0.04894	342.6	14.21	7.85	0.2620	24.96	54.18	79.14	104
105	1.1016	2.2429	0.05053	353.7	14.23	8.10	0.2627	25.20	55.95	81.15	105
106	1.1345	2.3099	0.05217	365.2	14.26	8.37	0.2635	25.44	57.79	83.23	106
107	1.1683	2.3786	0.05386	377.0	14.28	8.64	0.2642	25.68	59.69	85.37	107
108	1.2029	2.4491	0.05560	389.2	14.31	8.92	0.2650	25.92	61.64	87.56	108
109	1.2384	2.5214	0.05740	401.8	14.33	9.20	0.2658	26.16	63.66	89.82	109
110	1.2748	2.5955	0.05924	414.7	14.36	9.50	0.2667	26.40	65.73	92.13	110
111	1.3121	2.6715	0.06115	428.1	14.39	9.80	0.2675	26.64	67.87	94.51	111
112	1.3504	2.7494	0.06312	441.8	14.41	10.1	0.2684	26.88	70.08	96.96	112
113	1.3896	2.8293	0.06515	456.1	14.44	10.4	0.2693	27.12	72.37	99.49	113
114	1.4298	2.9111	0.06725	470.8	14.46	10.8	0.2703	27.36	74.72	102.08	114
115	1.4709	2.9948	0.06941	485.9	14.49	11.1	0.2712	27.60	77.16	104.76	115
116	1.5130	3.0806	0.07164	501.5	14.51	11.5	0.2722	27.84	79.66	107.50	116
117	1.5563	3.1687	0.07393	517.5	14.54	11.8	0.2733	28.08	82.24	110.32	117
118	1.6006	3.2589	0.07629	534.0	14.56	12.2	0.2743	28.32	84.89	113.21	118
119	1.6459	3.3512	0.07871	551.0	14.59	12.6	0.2754	28.56	87.62	116.18	119
120	1.6924	3.4458	0.08123	568.6	14.61	13.0	0.2766	28.80	90.46	119.26	120
121	1.7400	3.5427	0.08385	587.0	14.64	13.4	0.2777	29.04	93.42	122.46	121
122	1.7888	3.6420	0.08650	605.5	14.66	13.9	0.2789	29.28	96.40	125.68	122
123	1.8387	3.7436	0.08930	625.1	14.69	14.3	0.2802	29.52	99.56	129.08	123
124	1.8897	3.8475	0.09211	644.7	14.71	14.8	0.2815	29.76	102.73	132.49	124
125	1.9420	3.9539	0.09508	665.6	14.74	15.2	0.2828	30.00	106.09	136.09	125
126	1.9955	4.0629	0.09812	686.8	14.76	15.7	0.2842	30.24	109.52	139.76	126
127	2.0503	4.1745	0.1013	708.9	14.79	16.2	0.2856	30.48	113.08	143.56	127
128	2.1064	4.2887	0.1045	731.4	14.81	16.7	0.2870	30.72	116.70	147.42	128
129	2.1638	4.4055	0.1078	754.7	14.84	17.3	0.2885	30.96	120.47	151.43	129

The enthalpy of the liquid condensate is $h_o = t - 32$
$= 20$ Btu per lb.

By equation 5,

$q = (h_1 - h_2) - (w_1 - w_2)\, h_o$

$q = (38.198 - 23.154) - (0.01105 - 0.007637)\, 20$

$q = 15.044 - 0.06826$

$= 14.976$ Btu per lb of air, heat to be removed by the refrigerant.

Example 2: Humid air having the same initial wet bulb and dew point temperatures as in Example 1 is to be cooled to a

Properties of Mixtures of Air and Saturated Water Vapor (Continued)

Barometric Pressure 14.696 Lb per Sq In.

All tabulated values (except temperatures and pressures) are quantities per pound of dry air in the mixture.

Temp. Deg Fahr. t	Pressure: Lb per Sq In. p_v	Pressure: Inches of Mercury p_v	Absolute Humidity: Pounds w	Absolute Humidity: Grains w	Volume: Dry Air Cu Ft v_D	Volume: Volume Increase for Humid Air *Per Cent*	Humid Specific Heat s	Enthalpy (Heat Content): Dry Air Btu h_a	Enthalpy (Heat Content): Water Vapor Btu h_v	Enthalpy (Heat Content): Air-Vapor Mixture Btu h	Temp. Deg Fahr. t
1	2	3	4	5	6	7	8	9	10	11	12
130	2.2225	4.5251	0.1113	779.3	14.86	17.8	0.2901	31.20	124.46	155.66	130
131	2.2826	4.6474	0.1149	804.4	14.89	18.4	0.2917	31.44	128.50	159.94	131
132	2.3440	4.7725	0.1185	829.7	14.91	19.0	0.2933	31.68	132.60	164.28	132
133	2.4069	4.9005	0.1224	856.5	14.94	19.6	0.2951	31.92	136.93	168.85	133
134	2.4712	5.0314	0.1263	884.3	14.96	20.2	0.2969	32.16	141.43	173.59	134
135	2.5370	5.1653	0.1303	912.4	14.99	20.9	0.2987	32.40	145.97	178.37	135
136	2.6042	5.3022	0.1346	942.1	15.02	21.5	0.3006	32.64	150.77	183.41	136
137	2.6729	5.4421	0.1390	972.7	15.04	22.2	0.3025	32.88	155.75	188.63	137
138	2.7432	5.5852	0.1435	1004.4	15.07	23.0	0.3046	33.12	160.87	193.99	138
139	2.8151	5.7316	0.1481	1037.0	15.09	23.7	0.3067	33.36	166.15	199.51	139
140	2.8886	5.8812	0.1530	1070.7	15.12	24.5	0.3088	33.60	171.62	205.22	140
141	2.9637	6.0341	0.1579	1105.4	15.14	25.3	0.3111	33.84	177.25	211.09	141
142	3.0404	6.1903	0.1632	1142.1	15.17	26.1	0.3134	34.08	183.20	217.28	142
143	3.1188	6.3500	0.1684	1178.9	15.19	26.9	0.3158	34.32	189.17	223.49	143
144	3.1990	6.5132	0.1740	1217.8	15.22	27.8	0.3183	34.56	195.47	230.03	144
145	3.281	6.680	0.1797	1257.9	15.24	28.7	0.3209	34.80	202.00	236.80	145
146	3.365	6.850	0.1858	1300.3	15.27	29.7	0.3236	35.04	208.88	243.92	146
147	3.450	7.024	0.1920	1343.7	15.29	30.7	0.3264	35.28	215.94	251.22	147
148	3.537	7.202	0.1984	1388.5	15.32	31.7	0.3293	35.52	223.20	258.72	148
149	3.627	7.384	0.2050	1435.0	15.34	32.8	0.3323	35.76	230.77	266.53	149
150	3.718	7.569	0.2119	1483.5	15.37	33.9	0.3354	36.00	238.65	274.65	150
151	3.811	7.759	0.2191	1533.7	15.39	35.0	0.3386	36.24	246.82	283.06	151
152	3.906	7.952	0.2266	1586.1	15.42	36.2	0.3420	36.48	255.34	291.82	152
153	4.003	8.150	0.2343	1640.3	15.44	37.4	0.3455	36.72	264.16	300.88	153
154	4.102	8.351	0.2424	1696.9	15.47	38.7	0.3491	36.96	273.37	310.33	154
155	4.203	8.557	0.2508	1755.6	15.49	40.1	0.3529	37.20	282.93	320.13	155
156	4.306	8.767	0.2595	1816.7	15.52	41.4	0.3568	37.44	292.91	330.35	156
157	4.411	8.981	0.2686	1880.1	15.54	42.9	0.3609	37.68	303.24	340.92	157
158	4.519	9.200	0.2781	1946.7	15.57	44.4	0.3652	37.92	314.09	352.01	158
159	4.629	9.424	0.2880	2016.1	15.60	46.0	0.3696	38.16	325.40	363.56	159
160	4.741	9.652	0.2983	2088.2	15.62	47.6	0.3742	38.40	337.16	375.56	160
161	4.855	9.885	0.3091	2163.9	15.65	49.3	0.3791	38.64	349.50	388.14	161
162	4.971	10.122	0.3203	2242.4	15.67	51.1	0.3842	38.88	362.30	401.18	162
163	5.090	10.364	0.3321	2324.8	15.70	53.0	0.3895	39.12	375.75	414.87	163
164	5.212	10.611	0.3444	2410.8	15.72	55.0	0.3950	39.36	389.79	429.15	164
165	5.335	10.863	0.3573	2501.0	15.75	57.0	0.4008	39.60	404.51	444.11	165
166	5.461	11.120	0.3708	2595.3	15.77	59.1	0.4068	39.84	419.92	459.76	166
167	5.590	11.382	0.3849	2694.4	15.80	61.4	0.4132	40.08	436.11	476.19	167
168	5.721	11.649	0.3998	2798.6	15.82	63.7	0.4199	40.32	453.13	493.45	168
169	5.855	11.921	0.4154	2907.8	15.85	66.2	0.4269	40.56	470.98	511.54	169
170	5.992	12.199	0.4318	3022.6	15.87	68.8	0.4343	40.80	489.75	530.55	170
171	6.131	12.483	0.4490	3142.9	15.90	71.6	0.4421	41.04	509.42	550.46	171
172	6.273	12.772	0.4673	3271.1	15.92	74.5	0.4503	41.28	530.39	571.67	172
173	6.417	13.066	0.4864	3404.7	15.95	77.5	0.4589	41.52	552.25	593.77	173
174	6.565	13.366	0.5067	3547.2	15.97	80.7	0.4680	41.76	575.56	617.32	174
175	6.715	13.671	0.5281	3696.8	16.00	84.1	0.4777	42.00	600.05	642.05	175
176	6.868	13.983	0.5508	3855.9	16.02	87.7	0.4879	42.24	626.08	668.32	176
177	7.024	14.301	0.5748	4023.8	16.05	91.6	0.4987	42.48	653.58	696.06	177
178	7.183	14.625	0.6004	4202.6	16.07	95.6	0.5102	42.72	682.86	725.58	178
179	7.345	14.955	0.6275	4392.4	16.10	99.9	0.5224	42.96	713.90	756.86	179
180	7.510	15.291	0.6565	4595.3	16.12	104.5	0.5354	43.20	747.13	790.33	180
181	7.678	15.633	0.6873	4811.3	16.15	109.4	0.5493	43.44	782.53	825.97	181
182	7.850	15.982	0.7205	5043.5	16.17	114.7	0.5642	43.68	820.58	864.26	182
183	8.024	16.337	0.7558	5290.7	16.20	120.3	0.5801	43.92	861.11	905.03	183
184	8.202	16.699	0.7939	5557.2	16.23	126.3	0.5973	44.16	904.80	948.96	184
185	8.383	17.068	0.8349	5844.0	16.25	132.8	0.6157	44.40	951.82	996.22	185
186	8.567	17.443	0.8789	6152.6	16.28	139.8	0.6355	44.64	1002.4	1047.0	186
187	8.755	17.825	0.9267	6487.0	16.30	147.4	0.6570	44.88	1057.3	1102.2	187
188	8.946	18.214	0.9785	6849.7	16.33	155.6	0.6803	45.12	1116.8	1161.9	188
189	9.141	18.611	1.035	7245.7	16.35	164.6	0.7058	45.36	1181.8	1227.2	189
190	9.339	19.014	1.097	7677.6	16.38	174.3	0.7336	45.60	1252.5	1298.1	190
191	9.541	19.425	1.165	8152.2	16.40	185.1	0.7641	45.84	1330.4	1376.2	191
192	9.746	19.843	1.239	8673.0	16.43	196.9	0.7976	46.08	1415.9	1462.0	192
193	9.955	20.269	1.322	9252.6	16.45	210.0	0.8348	46.32	1511.1	1557.4	193
194	10.168	20.703	1.414	9895.2	16.48	224.6	0.8761	46.56	1616.6	1663.2	194
195	10.385	21.144	1.517	10617	16.50	240.9	0.9225	46.80	1735.1	1781.9	195
196	10.605	21.593	1.632	11427	16.53	259.2	0.9746	47.04	1868.1	1915.1	196
197	10.830	22.050	1.765	12352	16.55	280.1	1.034	47.28	2019.8	2067.1	197
198	11.058	22.515	1.915	13405	16.58	304.0	1.102	47.52	2192.9	2240.4	198
199	11.290	22.987	2.089	14620	16.60	331.5	1.180	47.76	2392.5	2440.3	199
200	11.526	23.467	2.292	16041	16.63	363.6	1.271	48.00	2625.9	2673.9	200

Vapor pressures from 32 to 200 F are reprinted by permission from "Thermodynamic Properties of Steam," by Keenan and Keyes, published by John Wiley & Sons, Inc.

final wet bulb temperature of 20 F and a dew point temperature of 15 F. Ice freezes on the coil surface and is cooled to a temperature of 12 F. Find the total heat to be removed by the refrigerant.

Solution:

$h_1 = 38.198$ Btu per lb

By equation 7,

$h_2 = 7.25 - (-149.05)\ (0.002290 - 0.001846)$

$= 7.25 + 0.0662 = 7.316$ Btu

(*Note:* At 20 deg, h'_{f2} in equation 7 is the enthalpy of ice and is found in column 2 of the table headed "Enthalpy of Ice and Low Temperature Water Vapor." At exactly 32 deg either the enthalpy of ice or of water should be used, depending on whether the air is in contact with ice or water.)

Equation 5 can now be used to compute the heat to be removed by the refrigerant.

From example 1, at 60 deg D.P., $w_1 = 0.01105$ lb.

From column 4, at 15 deg D.P., $w_2 = 0.001846$ lb.

From column 2 of the table headed "Enthalpy of Ice and Low Temperature Vapor," at 12 F,

$h_a = -152.85$ Btu per lb

By equation 5,

$q = (38.198 - 7.316) - (0.01105 - 0.001846)\ (-152.85)$

$= 30.882 + 1.407$

$= 32.289$ Btu per lb of air, heat to be removed by the refrigerant.

In the first example, the numerical value of the second term of equation 5 is negligible as it amounts to only 0.00826 Btu per lb. In the second example where ice formed, the numerical value of the second term is 1.407 Btu, or 4.4 per cent of the total. As a general rule, the second term may be neglected if there is no freezing of the condensate. However, if the moisture condensed from the air freezes, it is apparent from the second example that the second term is too large a proportion of the total to be neglected.

If freezing of the condensate does not take place, or if the weight of moisture condensed is very small, the second term of [5] may often be neglected, and this equation written as,

$$q = (h_1 - h_2) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [5a]$$

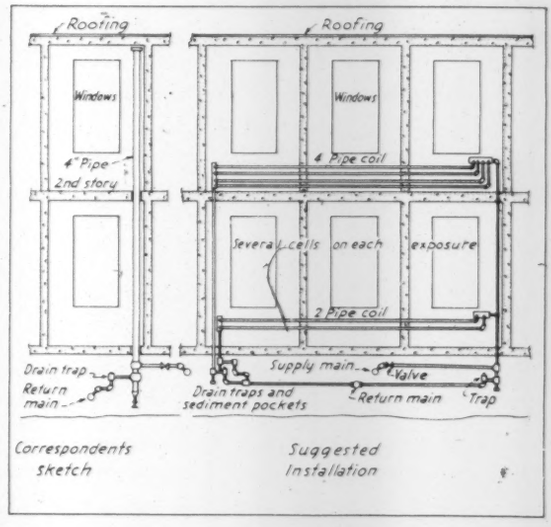
[A second article will discuss approximate methods of computing heat to be removed, compare the correct and approximate methods, and consider computation of absolute humidity, specific volume of dry air and air-vapor mixtures, and humid specific heat. The appendixes will show derivations of the equations, details of computations, etc.]

Novel Scheme Proposed for Heating Jail

A READER wishes a criticism of a novel scheme for heating cells in a two story jail. It is desired that no valves, traps, or other detachable apparatus shall be within reach of the prisoners and that even the radiator shall be substantially attached to the building structure. He suggests (see left-hand sketch) a 4 in. capped pipe running vertically through each pair of cells, with a ¾ in. valved steam supply connection in a tee in the pipe space under the lower cell and with a ½ in. thermostatic trap out of a tee in the same 4 in. vertical pipe just below the steam supply tee.

REPLY BY SAMUEL R. LEWIS*—It is clear that the heat loss from the first story cell will be less than that of the second story cell because the latter is subjected to the heat loss through the uninsulated roof. Therefore the first story cell will need less heating surface than the second story cell. In all probability, since air is heavier than steam, the 4 in. vertical pipe radiator will become hot throughout its length, though it is likely that the upper part in the second story cell will respond to the application of heat much more slowly than will the lower part in the first story. The first story requires approximately 12 sq ft of ordinary direct steam radiation. The pipe provided has 11.11 sq ft of surface, but since about half of this pipe is above the head level, it will not be as effective as would the same amount of surface in a conventional floor-located radiator. The second story requires about double the amount of radiation needed for the first story.

For these reasons I would not advise the method indicated for heating these cells. It seems probable that the heat distribution could be "horizontal" instead of "vertical," achieving the results desired without the objections (see right-hand sketch). For instance, while we do not have the complete plans, it is possible that a nearly horizontal pipe radiator could be run through the first story cells, one such coil, for instance, for each outside wall of the building. The valve and trap for each coil could be in the basement. The supply manifold could be in one cell and the return in another. The second story cells would be treated similarly, preferably with a separate valve and trap. Perhaps the control valves could be placed in the plumbing stack recess, accessible by the turnkey from the corridor, so as to give a better chance to regulate the heat, since the supply valve as shown in the shallow pipe space probably would be neglected.



It would be necessary, with the supply valves in the pipe space, to install traps on the supply riser bases beyond the valves, as indicated, so as to prevent water hammer when the valve is opened after having been closed.

I have shown the two horizontal pipe coils served by one supply valve. Separate valves, after the manner shown for the separate traps, would be better. A steam supply valve always should be at the high point so that when the valve is closed condensate can drain back to the supply main and on through the radiator. If the valve is below the radiator as indicated, an extra drain trap is required in order to prevent noisy operation.

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Modern Practice in Piping Design

The Steam and Boiler Feed Piping for the 865 Lb, 910 F Extension of Delray

WHEN the first section of Delray power house No. 3 was designed and built some ten years ago, throttle steam conditions of 400 lb gage and 700 F were adopted as suited to the then available materials and equipment designs. The possibility of using a higher steam pressure was given considerable thought at the time, but final selection was determined by the limitations of riveted boiler drum construction with respect to required plate thickness. The cost of seamless forged steel drums was prohibitive, while the welded drum construction now used almost universally for large power boilers was not countenanced in the A.S.M.E boiler construction code until some years later. Furthermore, the Company's engineers have a predilection for the simple regenerative turbine cycle without reheat, and a pressure of 400 lb approached the limit for a turbine operating without reheat at an initial temperature of 700 F which was then considered hot enough for available materials used in superheaters and turbines.

During the intervening years the Company has experimented with operation on a small scale at temperatures as high as 1000 to 1100 F, and has rebuilt its Conners Creek power plant for steam conditions of 600 lb and 825 F at the throttle, using welded boiler drums and alloy steel tubes in the last passes of the superheaters. Carbon steel pipe, of a superior grade obtained by "killing" or degasifying the molten steel by the addition of silicon and aluminum before pouring the ingots, was found satisfactory for temperatures up to 850 F, and trouble from leaky pipe joints was avoided by going to strength-welded construction.

Early in 1937, however, when increased electrical load called for an extension of the Delray plant, turbine builders and superheater manufacturers had progressed to the point where they were prepared to offer equipment for temperatures of 925 to 950 F. This allowed a corresponding increase in throttle pressure without producing excessive moisture in the lower stages of the turbine, a consideration which is a criterion of design on account of the tendency for erosion of the blading in these stages in case the moisture content of the exhaust exceeds 10 to 15 per cent. Moisture restrictions are sufficiently elastic, however, to allow a considerable choice in combination of throttle pressures and temperatures within these limits, ranging from say 800 lb at 900 F to 1250 lb at 950 F. While a number of "topping" units for pressures of around 1200 lb and temperatures of about 925 F have been superposed on old low pressure turbines by other companies, this choice of pressure and temperature is not necessarily the most economical from all angles for wholly new equipment divested of the need for matching the high pressure end to an existing low pressure unit.

Analysis of turbine heat cycles for the proposed Delray extension indicated a substantial fuel saving in going from 600 lb, 825 F to around 800 lb, 900 F, with a relatively small fuel saving in stepping up from 800 lb, 900 F to 1200 lb, 900 F or thereabouts. Assuming 45 per cent plant use factor, 12½ per cent fixed charges, and $4.00 coal, the first step would justify an additional plant outlay of about $2.75 per kilowatt, as against only 75 cents per kilowatt for the second step. Obviously with a new unit there was little incentive under present conditions to go much beyond the 800 lb, 900 F step. Hence, after some consideration of manufacturers' proposals, decision was made to design for the following schedule of steam conditions:

SCHEDULE OF STEAM CONDITIONS

Location	Pressure Lb per Sq In. Gage	Temperature Deg F
Superheater Outlet		
Normal	865	910
Maximum	900	925
Turbine Throttle		
Normal	815	900
Maximum	850	910
Piping Design		
Continuous	865	910
Short Periods	900	925

Distinction between the "continuous" and "short period" figures for piping design is made to permit consideration of short time bursting pressures on the one basis, while creep or plastic flow is treated on a continuous long time basis. Furthermore, in ordering valves and other equipment which might be damaged by excessive temperature or pressure, it is advisable to specify that they be capable of withstanding the maximum conditions, even though expected for only short periods of time. This need is emphasized by the rapid falling off in strength of some materials for comparatively small increases in temperature above 900 F.

The size turbo-generator decided on for the extension is 75,000 kw. While this capacity is stepped up considerably from the 50,000 and 60,000 kw units which have been the largest heretofore installed on the Detroit Edison system, the increase scarcely matches the electrical load growth so that no larger percentage of the total load is "carried in one basket". The turbine house at the Delray plant originally was intended for six 50,000 kw turbo-generators, operating on 400 lb, 700 F steam and turning at 1200 rpm. Three of these units already have been installed. The space intended for the other three fifties will nicely accommodate three seventy-fives operating on 815 lb, 900 F steam and turning at 1800 rpm, since the higher steam pressure and speed tend toward more compact units. One seventy-five will be installed ready for operation in the fall of 1938 and another the following year, while the third will not be scheduled at this time pending observation of the growth in load for the plant area. In the meanwhile a 10,000 kw, 815 lb, 900 F topping unit, using an existing gener-

ator and requiring no extension of the building, will be installed to make available to the 400 lb, 700 F units up to 500,000 lb per hr of high pressure steam, thus reducing coal consumption and at the same time compensating for some deficiency of reserve capacity in the 400 lb boilers.

Choice of Materials for 900 to 925 F

In going to operating temperatures above 850 F, consideration must be given to choice of mild alloy steels for pipe, valve bodies, and flanges having high temperature properties superior to carbon steel. For temperatures below 950 F, carbon-molybdenum steel has met with much favor, both for seamless pipe and for cast valve bodies and fittings.

*By Sabin Crocker**

Here is a thorough discussion of modern practice in the design and layout of power plant piping for high pressures and temperatures, using the new extension for the Delray plant in Detroit as the example . . . Mr. Crocker, who is widely known for his piping standardization activities, explains in detail the piping problems in a project of this kind, and how they are being met at Delray

Pipe

The relatively high cost of carbon-molybdenum pipe (*viz.*, two to three times that of plain carbon steel of the "killed" variety) has precluded its use at temperatures below 850 F. However, a comparison of creep test results[1] on carbon-molybdenum and killed carbon steel at 910 F indicates that a working stress approximately 50 per cent greater can be assigned the carbon-moly material at that temperature. Or, stating the proposition another way: carbon-moly has almost as great creep strength at 910 F as carbon steel has at 750 F, and about 15 per cent greater creep strength at 910 F than carbon steel has at 850 F. These favorable relations, which are supported by the recommended working stress curves of Fig. 1 (plotted from data in proposed Table P-8 of the A. S. M. E. boiler code as published in *Mechanical Engineering*, June, 1936) made it possible to use at Delray Schedule 100 carbon-moly pipe, which is only 20 per cent thicker than the Schedule 80 killed carbon steel pipe used at Conners Creek, despite a 40 per cent increase in steam pressure.

The carbon-moly pipe, which is sometimes referred to as ½ per cent molybdenum steel, was bought to conform to A. S. T. M. Specification A206 calling for the following chemical composition and room temperature physical properties in the normalized and drawn condition:

Chemical Composition	
Carbon, per cent	0.10 to 0.20
Manganese, per cent	0.30 to 0.60
Phosphorus, per cent	0.04 max
Sulphur, per cent	0.05 max
Silicon, per cent	0.10 to 0.25
Molybdenum, per cent	0.45 to 0.65
Physical Properties, Minimum	
Tensile strength, lb per sq in.	55,000
Yield point, lb per sq in.	30,000
Elongation in 2 in., longitudinal, per cent	30
Elongation in 2 in., transverse, per cent	25

For an internal pressure of 865 lb, the computed value of the stress which tends to resist bursting of the pipe, taking into account the reduction in wall thickness of 0.065 in. for corrosion as required by the Code for Pressure Piping, amounts to 8250 lb per sq in. for the 10 in. pipe size, and slightly less for the 8 and 12 in. sizes. The corresponding stress for 10 in. pipe based on the minimum wall thickness permitted on inspection (before taking out the corrosion allowance) is only 7400 lb per sq in. The stress of 8250 lb at 910 F chances to fall almost exactly on the boiler code committee's allowable stresses for carbon-moly plate as plotted in Fig. 1. Somewhat higher allowable stress values for carbon-moly plate and pipe materials are now under consideration by this committee.

On the strength of the foregoing relation, decision was made to use Schedule 100 carbon-moly pipe for the superheated steam piping in the Delray extension in sizes 8 in. and larger, and to use the next heavier Schedule 120 for sizes 6 in. and smaller, since Schedule 100 has no sizes under 8 in. The resulting selections are in accordance with those agreed to in 1936 by the Prime Movers Committee of the Edison Electric Institute and the pipe manufacturers for stocking carbon-moly pipe suitable for nominal conditions up to 900 lb at 900 F.

Incidentally, Schedule 140 carbon-moly pipe was used for the steam lines to the 1200 lb, 900 F unit at Ford's Rouge plant and for the first section of the Waterside No. 2 plant of the Consolidated Edison Co. of New York. If these plants have a pressure drop of 50 psi (psi = pounds per square inch) from the superheater outlet to the turbine, the working stress by the Code for Pressure Piping formula would be 8300 psi for the 10 in. size and 8660 psi for the 12 in., which stresses chance to agree closely with the Delray design.

Valves

The practice of welding valves into the line to eliminate flanged joints calls for casting valve bodies of a mild alloy steel which has good welding properties and does not tend to air harden excessively. The necessity for tapering down the wall thickness at the welding ends to approach that of the pipe calls for a cast alloy steel body possessing high temperature physical properties commensurate with those of the pipe material and at the same time having a soundness in the cast structure of the welding end which approaches that in the wrought structure of the pipe. Furthermore, on account of the reduced section at the welding ends and the requirement of strength welding, the valve material should possess a high ability

*Engineer, The Detroit Edison Company. Member of Board of Consulting and Contributing Editors.

[1] "Properties of Steel Pipe and Tubes Applicable to High Temperature and High Pressure Service", National Tube Co., 1936.

to resist shock as demonstrated through impact strength tests.

Since the significance of impact tests is not any too well understood, it may be helpful to digress briefly on just what the results of Charpy or Izod tests on specimens from valve bodies are expected to show. Impact strength is more of an indication of soundness of structure and of carrying on manufacture at proper temperatures than it is of the strength or ductility of the material. To the metallurgist its chief value is to provide a check by which to detect material which has been overheated or inadequately annealed, or which possesses temper brittleness, aging embrittlement, porosity, or other conditions which are known to be detrimental. Because the notched bar impact value often is unique in indicating a condition which might cause material to fail under shock, it is deemed of unusual significance in the inspection of cast steel used in welding end valve bodies.

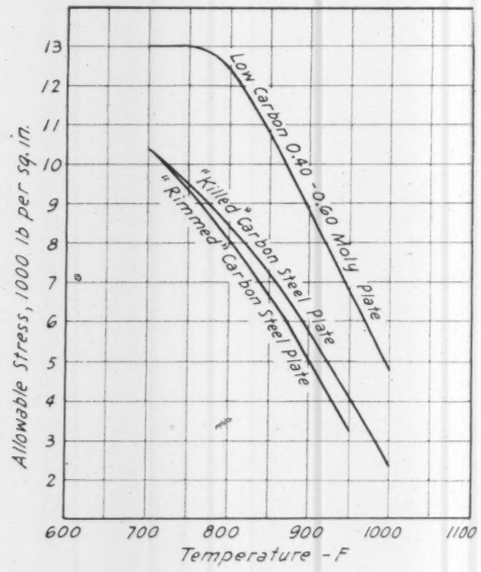


Fig. 1—Allowable stresses at elevated temperatures for carbon steel and carbon-molybdenum steel plate, plotted from values recommended by the subcommittee on ferrous materials of the A. S. M. E. boiler code committee

While carbon-molybdenum cast steel was considered entirely satisfactory for valve bodies in steam lines operating at temperatures of 900 to 950 F, the Detroit Edison Company has had exceptionally good results in the 850 F piping at its Conners Creek plant with a nickel-chrome-molybdenum alloy of the following composition and physical properties:

COMPOSITION AND PHYSICAL PROPERTIES OF NICKEL-CHROME-MOLYBDENUM CAST STEEL

CHEMICAL COMPOSITION

Carbon, per cent	0.28 max
Manganese, per cent	0.70 max
Nickel, per cent	1.05 max
Chromium, per cent	0.70 max
Molybdenum, per cent	0.40 max
Silicon, per cent	0.25 min
Phosphorus, per cent	0.05 max
Sulphur, per cent	0.05 max

PHYSICAL PROPERTIES (MIN)

Tensile strength, lb per sq in.	85,000
Yield point, lb per sq in.	60,000
Elongation in 2 in., per cent	20
Reduction of area, per cent	35
Charpy V-notch, ft-lb	30

Due to the higher alloy content of this material, both its room temperature physical properties and its creep resistance at the operating temperature are somewhat superior to carbon-molybdenum steel. The rather unusual practice of specifying only the maximum percentage for several of the alloy contents, including carbon, manganese, nickel, chromium and molybdenum, was dictated by the necessity of reducing the tendency of the material to air harden when welded. On the other hand, the addition in the foundry of sufficient alloy content is insured through the necessity for meeting rather stiff physical tests on specimens cut from coupons cast with each valve body. In view of the satisfactory performance of this material at Conners Creek, it was selected for the high temperature valve bodies in the Delray addition.

Precautions in Welding Alloy Steels

The Company's carefully developed technique in welding carbon steel pipe at Conners Creek in pipe-to-pipe and pipe-to-alloy-valve-body joints has been described at length in previous articles.[2] This technique involves rigorous qualification tests, and the use of coated electrodes, the arcronograph, and an electric induction stress reliever. While the welding at Conners Creek does involve joining carbon steel pipe to carbon-moly and nickel-chrome-moly valve bodies, it does not require joining two alloy steel members together with a welding rod having molybdenum in its coating. The latter operation is somewhat more difficult since there seems to be a tendency for the fused metal to air harden unless the joint is preheated and held at a temperature of 300 to 400 F or higher throughout the welding operation.

Molybdenum is said to give steel an air hardening tendency when present in amounts of the order of ½ to ¾ of 1 per cent, which percentage overlaps with the permissible range of molybdenum in carbon-moly pipe. It is possible that the high melting point of molybdenum, nearly 2000 deg F higher than iron and most of its alloying elements, has something to do with this, especially since there is considerable molybdenum in the electrode coatings used for such work. This comes about through the necessity for keeping the composition of fused metal in the weld on a par with that of the parts being joined, the means of accomplishing this at present in favor being to use a carbon steel welding rod having the required molybdenum in its coating.

Where preheating is not employed in welding carbon-moly pipe there is said to be a tendency to form fine cracks in the weld metal before it can be stress relieved. This tendency is particularly pronounced in the layers first deposited, since the chilling effect is more severe at the outset of welding before the arc itself has had a chance to heat the joint. Examinations for hardness value of cross sections of preheated and unpreheated welds of carbon-moly pipe show greater variations and higher hardness values throughout the unpreheated specimens. Such variation in hardness value is considered

[2]HEATING, PIPING AND AIR CONDITIONING, February and March, 1935, and January, 1937.

undesirable, since it is thought to indicate a condition tending toward easier fracture by shock than where uniform hardness obtains.

For the reasons stated above, and until some simpler technique is developed, it was decided to preheat all welds at Delray where carbon-moly pipe is involved, and to maintain a metal temperature at the joint in excess of 300 F until welding is completed.

Choice of Dimensional Standards for 865 Lb, 910 F

The pressure and temperature ratings given in the present American Standards for Steel Flanged Fittings and Companion Flanges, A. S. A. B16e-1932, are for carbon steel. Since it is common practice to employ alloy castings of one kind or another for valve bodies, tables of ratings for carbon-moly or equivalent alloy are now under consideration. Two distinct tables of ratings have been approved by Subcommittee 3 of A. S. A. Sectional Committee B16. The table for valves having raised face end flanges rates the American Standard for 900 lb S. S. P. as good for 900 lb at 900 F if made of carbon-moly or equivalent alloy material. A second table based on the use of ring type end flanges proposes to rate the present 900 lb standard as good for 900 lb at 950 F when made of alloy material.

On this same basis even a higher rating might reasonably be assigned to welding end valves, as the problem of keeping bonnet joints tight is not complicated by bending moments as is true in the case of end flanges.

The metal thickness of the 900 lb standard is adequate for the higher temperature rating proposed for carbon-moly. For example, the stress in the cylindrical portion of a 10 in. valve body based on the minimum metal thickness is only about 4300 psi by the straight Barlow formula for a pressure of 900 psi. Even if this stress were increased 50 per cent by reason of the bonnet opening, as is customarily assumed, it would be less than 6500 psi.

The bursting stress in the reduced section of a welding end valve turned down to $\frac{1}{16}$ in. greater than the nominal wall thickness of Schedule 100 pipe is approximately 6300 psi for an internal pressure of 900 psi. There would be no advantage in using a heavier valve standard since the welding end would be the same thickness in any case.

On the basis of the proposed ratings for alloy flanged valves, a check of the minimum wall thickness, and calculations of the effect of bending moments, the following standards were selected for 865 psi, 910 F service:

(1) Welding-end valves; Amer. Std. 900 lb. S.S.P.
(2) Flanged-end valves; Amer. Std. 1500 lb S.S.P.
(3) Superheater safety valves; Amer. Std. 1500 lb S.S.P.

In the specifications for boilers the connections for superheater safety valves were required to conform to American Standard 1500 lb dimensions. While, in the absence of bending moments, the above reasoning would demonstrate that 900 lb flanges should be adequate for these connections, quotations showed no cost advantage in using the lighter standard. This condition arose from the fact that since only 1500 lb patterns were available, 1500 lb bodies would have been supplied with either weight of end flange.

For the superheater safety valve connections and for all other flanged end valves, decision was made to use a modified sargol joint with a ground face, but not to seal weld the lips unless found necessary on trial to insure tightness. Since it appeared that a gasket for 910 F service should be of metal, hydrogen annealed soft furniture iron was adopted as the most promising. These gaskets will be $\frac{1}{32}$ in. thick by $\frac{3}{8}$ in. wide, with a smooth contact surface.

The main steam valves are of the parallel slide type with welded in seat rings and stellite facings on seats and discs. Motor operators are provided for closing any important valve from a remote control station in case of emergency. Bonnet joints are of the male and female type using monel metal gaskets and Grade B13 (A. S. T. M. A-193-37T) alloy steel bolts.

Effect of Thermal Expansion

Provision for absorbing thermal expansion in 865 lb, 910 F piping is something of a problem due both to the sizeable elongation and to the increased rigidity resulting from the use of thicker walled pipe. The coefficient of expansion for carbon-molybdenum steel is not much different from that of carbon steel, giving an increase in length of about 8 in. per 100 ft in going from atmospheric temperature to the normal operating temperature of 910 F. Normal steam conditions for continuous operation, rather than maximum conditions expected for short periods only, were made the basis of design for flexibility since it was felt that disregarding the possible benefits obtained through cold springing the line during erection more than offset the minor differences involved.

The following combined longitudinal pressure-plus-bending stresses were adopted as suitable for carbon-moly pipe for 910 F service. These stresses are the same as were used at Conners Creek for killed carbon steel pipe at 850 F except that pipe wall stresses adjacent to welded joints were stepped up from 10,000 to 12,000 psi.

Smooth bends and straight pipe remote from joints	15,000 psi
Pipe wall adjacent to bolted line joints	10,000 psi
Pipe wall adjacent to welded line joints	12,000 psi
Pipe wall adjacent to creased bends or corrugated pipe	10,000 psi

Limiting the combined stress at bolted joints to 10,000 psi is deemed necessary to keep bending moments low enough to insure tight joints even when using 1500 lb flanges. In the case of welded joints there are less definite reasons for limiting the stress to 10,000 psi, and it was decided that a higher stress should be permitted at Delray. The original limitation to 10,000 psi at Conners Creek was made because of uncertainty then as to the quality of the cast welding ends on the valves and inexperience with welding. Subsequent investigation of welding end valves has demonstrated that sound metal can be expected in the welding ends and that welding as practiced in The Detroit Edison Company is entirely reliable. Increasing the combined stress at welded joints to 12,000 psi tended to offset the greater stiffness of thicker walled pipe and assisted materially in securing sufficiently flexible piping layouts.

While raised face, alloy steel flanges of the 900 lb standard apparently are considered satisfactory by A. S. A. Sectional Committee B16 for 900 lb at 900 F, there is considerable reason to question whether a gasketed joint could be maintained tight under this pressure and temperature condition if there were an appreciable

bending moment acting at the joint. Computations based on rather arbitrary assumptions of points of application of bolt loads and gasket reactions show that a bending moment corresponding to a total stress (longitudinal pressure-plus-bending stress) of 9000 psi in the pipe adjacent to a 12 in. joint probably would cause a 900 lb flanged joint to leak if used on 865 lb pressure and 910 F service.

Exactly similar calculations for a 1500 lb flanged joint show that it could be expected to maintain a tight joint indefinitely on 865 lb, 910 F service with a bending moment corresponding to a total stress of 10,000 psi in the pipe. These calculations were based on the assumption that carbon-moly or nickel-chromium-moly flanges would support a stress of 8000 psi at 910 F, while the chromium-tungsten-moly, Grade B13, alloy bolting material would carry 10,000 psi after stress adjustment with time had taken place.

Where there is no bending moment acting, as in the case of valve bonnet joints or connections to safety valves, the 900 lb standard should prove adequate for 865 lb, 910 F service. The residual gasket compression on a ⅜ in. wide gasket was determined as about 10 times the internal pressure for this condition.

Boiler Feed Piping

The boiler feed piping is designed to meet the following operating conditions at the boiler feed pump discharge:

	Pressure	Temperature
Normal (continuous) maximum boiler rating	1092 psi	Max. turbine load, leaving pump = 260 F
Maximum (short periods) maximum evaporation, all safety valves blowing (6 per cent over pressure)	1186 psi	Leaving last heater = 388 F
Shut-off pressure, maximum boiler rating, normal drum pressure	1325 psi	

Pipe

The superior grade of medium carbon steel pipe obtained for Conners Creek suggested its use for boiler feed piping at Delray. This special analysis, silicon killed, medium carbon steel possesses room temperature physical properties approaching that of Grade C pipe of A. S. T. M. Specification A106, although specified to conform to Grade B requirements.

The allowable stresses for Grade B and Grade C pipe as given in the power piping section of the Code for Pressure Piping for different working temperatures are plotted on Fig. 2.

Schedule 80 pipe, if made of material conforming to Grade B requirements, was deemed entirely adequate for the boiler feed pressure and temperature conditions. The hoop stress in 8 in. Schedule 80 pipe at maximum boiler rating is 12,660 psi as computed by the Code for Pressure Piping formula. The stress based on minimum wall thickness permitted on inspection is only 10,800 psi for this condition. The allowable stress for Grade B pipe for a temperature of 388 F, which is the maximum temperature leaving the last stage heater, is approximately 15,000 psi. For the temperature of 260 F at the discharge of the boiler feed pump the allowable stress is about 15,500 psi.

The stress in the pipe by the Code for Pressure Piping formula for the condition of maximum evaporation with allowance for 6 per cent drum over-pressure is 13,720 psi. Under shut-off pressure, maximum boiler rating, and normal drum pressure, the stress might rise to 15,400 psi by the code formula, or an actual stress of approximately 13,100 psi.

Even if a water hammer allowance of 600 lb for excess pressure were made the pipe would still have a factor of safety of about three, based on the yield strength of the pipe material as actually obtained for Conners Creek.

It has seemed necessary to give a rather complete analysis of the stress conditions in the pipe selected for boiler feed service at Delray since the increase in nominal pressure at turbine throttle of 215 psi over that at Conners Creek would naturally suggest that a heavier pipe schedule should be selected than was used at Conners Creek.

Valves

On the basis of the pressure and temperature ratings for carbon steel flanged fittings and companion flanges of A. S. A. B16e as amended in 1935, the 900 lb standard is adequate for boiler feed service at Delray. These ratings for the temperature range involved in the boiler feed system are tabulated below:

Service Temp F	Service Pressure 900 Lb Std.
250	1350
300	1305
350	1260
400	1215

Under normal maximum boiler rating the pressure at the boiler feed pump discharge will not exceed 1092 psi.

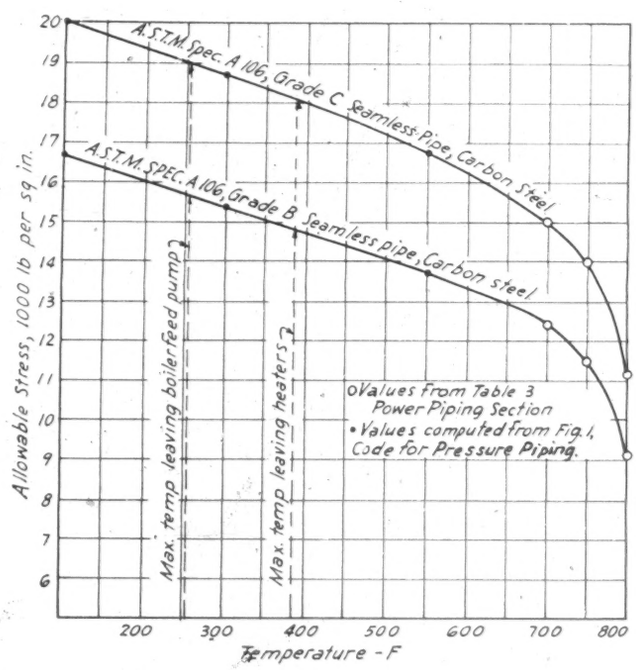
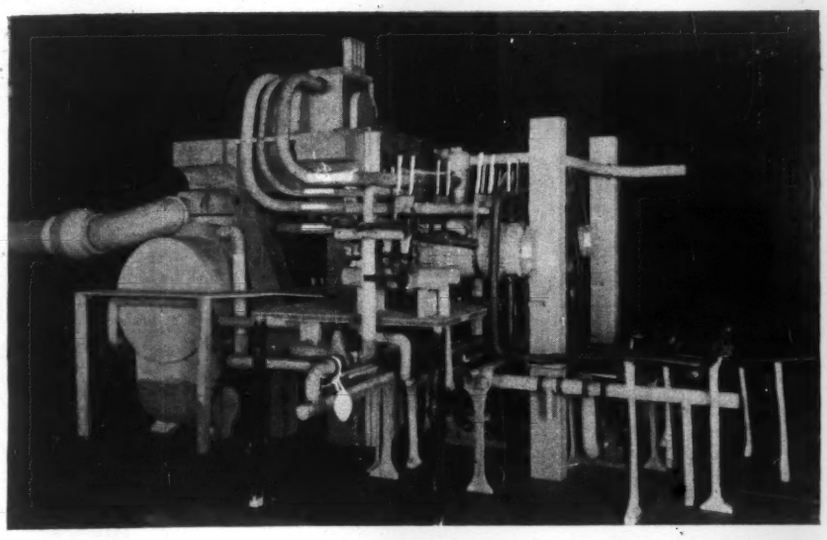


Fig 2—Allowable stresses at elevated temperatures for Grade B and Grade C seamless pipe, A. S. T. M. Specification A106, plotted from values given for power piping systems in A. S. A. Code for Pressure Piping, B31.1-1935



Scale models are used in plant design by The Detroit Edison Company with a view to: (1) eliminating interference with the building structure and equipment; (2) providing sufficient head room and dismantling space; (3) obtaining good appearance and operating convenience. Models are progressively revised as the design advances and before working plans of the various drafting divisions are approved until a generally satisfactory layout is assured. The model shown is for part of the Delray extension

Even with allowance of 6 per cent over pressure at maximum evaporation the pressure will be less than 1186 psi. The shut-off pressure for maximum boiler rating and normal drum pressure is expected to be about 1325 psi, or slightly above the rated working pressure of the American 900 lb standard at the maximum operating temperature of 388 F leaving the last stage heater.

These ratings are of necessity quite arbitrary but they do give an indication of past successful practice. There is no question of ability to secure and maintain adequate compression on the gasket surfaces with American 900 lb standard flanges on boiler feed service at Delray.

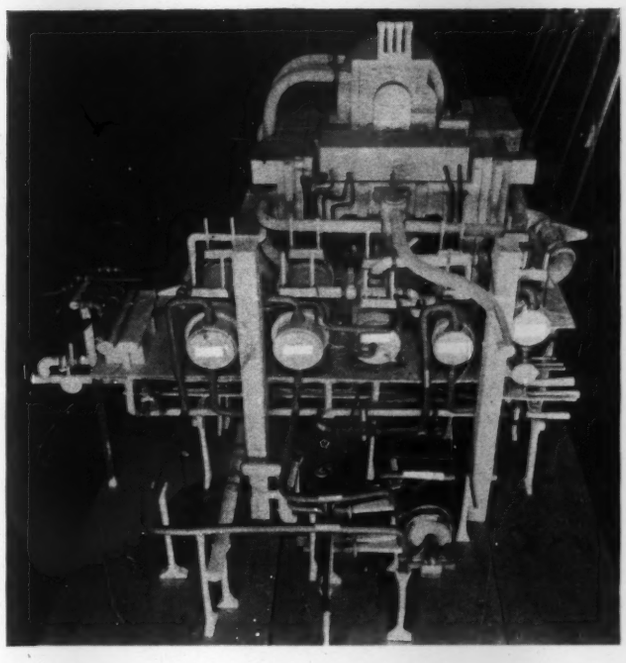
Since welding end valves have their ends turned down to $\frac{1}{16}$ in. greater than the nominal wall thickness of adjoining pipe, there is no advantage as far as the strength of the welding end is concerned in using valves heavier than the American 900 lb standard.

The terminating flanges on feedwater connections to the economizer are required by the boiler code to conform to the American 1500 lb standard. The valves connecting to the economizers are likewise required to conform to the American standard for 1500 lb. It was decided to have the remaining valves and fittings in the boiler feed system conform to the American standard for 900 lb.

Based on experience at Conners Creek, long fiber asbestos gaskets should prove satisfactory for feedwater connections. It is proposed to use ¼ in. raised face joints with serrated finish on the 1500 lb flanges connecting to the economizer and on the 900 lb flanges on boiler feed pump discharge, heater water boxes and other flanged joints in the boiler feed system. The remaining joints will be welded, including most of the valves. Both the 900 lb and 1500 lb gate valves used in the boiler feed system will be of the wedge type faced with No. 6 stellite on seats and discs. Similar facing will be used in globes and checks.

Piping Layout

The main steam and boiler feed piping systems of the Delray extension resemble in principle the layouts used in rebuilding Conners Creek. Both layouts differ in two respects from the corresponding piping used in the 400 lb section of the Delray plant, which was built between 1927 and 1930. First, the new main steam piping has been restricted somewhat to partially approach the unit boiler-turbine plan; and second, the new boiler feed piping has been curtailed from the complete dual system formerly used. These departures are explained below.



Main Steam

The differences in layout of main steam piping between the old and new sections at Delray are displayed in Fig. 3. In the old section it was deemed essential to have a 16 in. ring header, situated in the upper part of the boiler house, into which all boiler outlets are connected and from which all turbine leads take off. Through the somewhat devious paths of this ring header, the steam output of each boiler has available two distinct routes for reaching the take-off point to any turbine, both routes being of ample capacity to afford minimized pressure drops. This layout provides crosswise piping of a diameter larger than necessary if the direct leads from

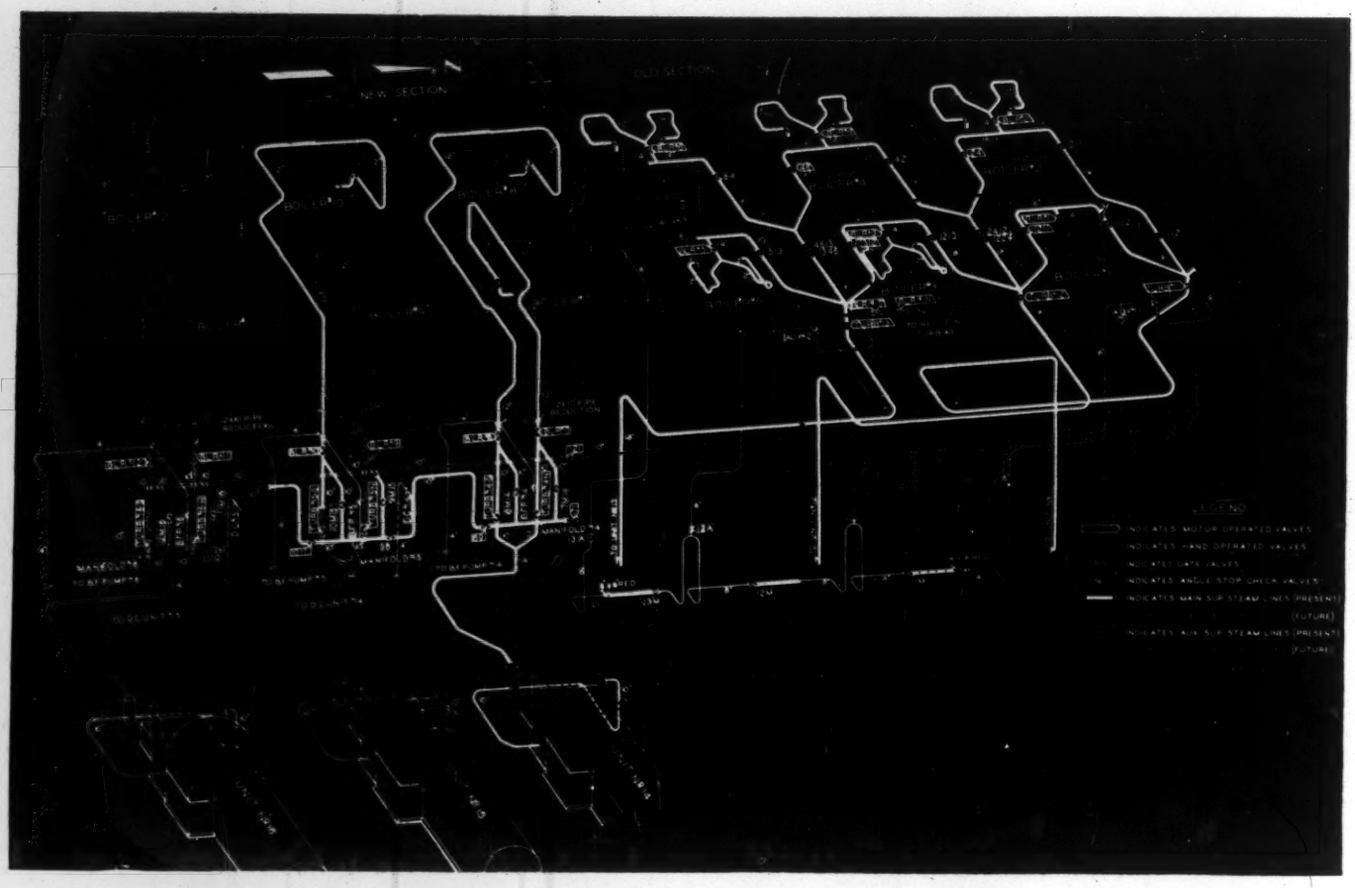


Fig. 3—Comparison of main steam piping in old and new sections of Delray power plant

boiler to turbine can be utilized the great part of the time as is expected in the so-called "unit boiler-turbine" plan. In this plan, used in several recent central stations, advantage is taken of the greater reliability inherent in present day boilers to eliminate cross-overs and restrict the main steam (and sometimes the boiler feed piping) to a single line between the turbine and its particular boiler or pair of boilers.

The new main steam system of Fig. 3 shows a partial approach to the unit plan in that each pair of high pressure boilers is direct connected to a 75,000 kw turbo-generator. Steam from any boiler may be bypassed, however, through a modest 10 in. ring header which serves to transfer steam crosswise of the plant when occasion requires. While some reduction in throttle pressure will ensue under full turbine load when steam is furnished from more distant boilers, such a condition would be unusual and the increased fuel consumption of small consequence because of its short duration.

The moderate extra cost of a small ring header over none at all is felt to be justified on the grounds of the greater availability of the various boiler-turbine combinations. The main steam systems installed at Conners Creek and the Delray extension are believed to combine the advantages of low pressure drop and general simplicity of the unit boiler-turbine system with the greater availability of the interconnecting header system. An added advantage of this system has been the ability to hold down to 10 in. the size of the parallel slide gates used for steam stop valves.

Boiler and turbine stop valves, and header division valves, are motor operated from a remote control station near the plant engineer's office. Valves are closed electrically, but in accordance with established practice of the Company, they must be opened by hand. Opening these valves by hand is considered desirable so that the operators can observe the pressure gages on both sides to insure equalizing the pressure when placing a line in service. This also tends to reduce the possibility of opening a valve by mistake.

Boiler Feed

The boiler feed system for the new section of Delray shown in Fig. 4 consists of a single ring header fed through one riser from the pumps of each turbine unit which connects to the ring header between each battery of boilers. Sectionalizing valves are provided which, combined with the ring header, enable water to be fed any boiler in the event of piping difficulty in a riser or in a portion of the header normally used to supply that boiler.

The Company's practice prior to rebuilding Conners Creek required full duplicate systems of feedwater piping, including dual risers from the pumps and dual headers (not ring headers) having separate connections to the boilers. An extensive study of the reliability of boiler feed systems in the Company's older plants demonstrated that since no real emergency had occurred to require the use of a dual supply system, such extensive duplication could no longer be considered essential. Furthermore, the need for a complete auxiliary feed system has become less evident as the dependability of piping construction has increased through the use of welded joints and better materials.

On the other hand, due to the high degree of reliability expected of a boiler feed system, it was thought unwise

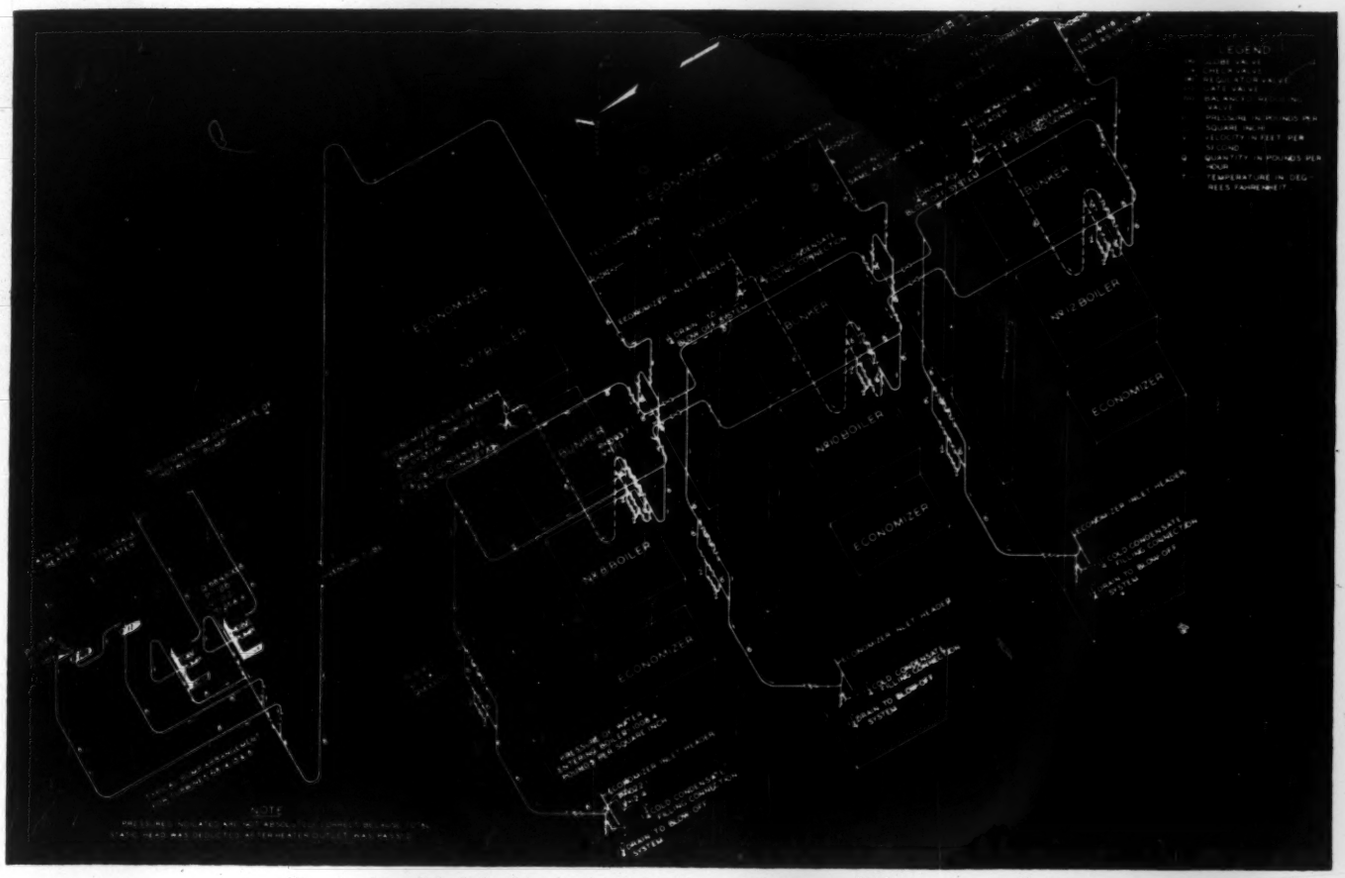


Fig. 4—Simplified boiler feed piping in extension to Delray power plant

to depend entirely on a single line of supply all the way from pump to boiler. Hence, in rebuilding Conners Creek, it was deemed a good compromise between economy and reliability to substitute a single ring header near boiler drum elevation in lieu of the dual headers formerly used at this point, to use one branch from this header to the boiler, and to eliminate the auxiliary riser from the pumps to this elevation. Satisfactory results at Conners Creek have called for repeating with essentially the same arrangement, shown in Fig. 4, in the new section at Delray. In addition to eliminating a considerable amount of parallel piping, this simplification effected a notable reduction in the number of valves required, both at the pump discharge and the boiler drums. An incidental advantage in eliminating the auxiliary feed system is the saving in heat loss chargeable to keeping the emergency line hot, as is done in the older plants by continuously bleeding a small amount of hot water back to storage.

The boiler feed risers and ring headers were designed for a water velocity of 10 ft per second under normal maximum flow conditions, which was considered about the top velocity consistent with safe design. An investigation at one of the Company's other plants of water hammer actually produced at different water velocities through sudden closure of a valve or tripping out of a boiler feed pump, showed that velocities in excess of about 10 fps produced shock pressures greater than deemed permissible without increasing the pressure rating standard of the line. While velocities up to 15 or even 18 fps appear from numerous studies to be economically justified from the standpoint of pumping cost, better methods for minimizing the effect of water hammer are required before such velocities are used.

Erection of piping at Delray is under way as this article is published. Most of the technique developed in connection with the welded piping at Conners Creek is being carried over into Delray and a few extra precautions taken on account of using carbon-moly pipe for superheated steam as described under "Precautions in Welding Alloy Steels." Details of welding grooves and backing rings for butt welded joints in sizes 2½ in. and larger, and for socket fittings and fillet welds in the smaller sizes, are essentially as evolved at Conners Creek where they were found to work out well. Also the methods for attaching welded branch outlets to manifolds, inserting welded in flow nozzles, and attaching small drain and instrument lines to larger pipes is unchanged.

Experience in welding high pressure joints at Conners Creek has confirmed the Company's practice of using the arcronograph as a check on the uniformity with which weld metal is deposited. Likewise, use of the electric induction heater developed for stress relieving welds in thick walled pipe at Conners Creek will be continued at Delray. Preliminary investigation of x-ray methods for examining field welds, however, has not so far disclosed a sufficient degree of accuracy in locating artificially induced defects in test welds to justify extensive use of this device. Until greater dependence can be placed on the findings of non-destructive examinations of field welds made without taking a prohibitive number of shots, the company's engineers prefer to rely on the graphical record of the arcronograph as a simple device for obtaining quality control in depositing weld metal.

Air Conditioning Design and Operation Demand Real Engineering Skill

By H. Warr*

IT has been said that to be successful the department store must have the merchandise that people want to buy at prices they are willing to pay. Fundamentally, this cannot be disputed; but we may add many other factors that have direct bearing, such as appealing display of the merchandise, adequate illumination, adequate elevator service, cleanliness that appeals to the public and avoids spoilage of merchandise, and carefully trained and efficient sales people.

Since the temperature and relative humidity of the air have much to do with our feeling of comfort or discomfort, we can justly consider that air conditioning also plays an important rôle and, therefore, is a factor to consider in the operation of a department store. Shoppers demand and should have comfort. The store attendants and sales people are more efficient, more cheerful and less subject to colds and other ailments which keep them away from work, when they are comfortable. The store atmosphere should always be clean and fresh, and should be maintained at suitable conditions of temperature and relative humidity. These are some of the facts that appealed to the sound economic reasoning of J. L. Hudson Co. officials who, as pioneers of progress, were among the first to adopt the science of real air conditioning as applied to department store operation.

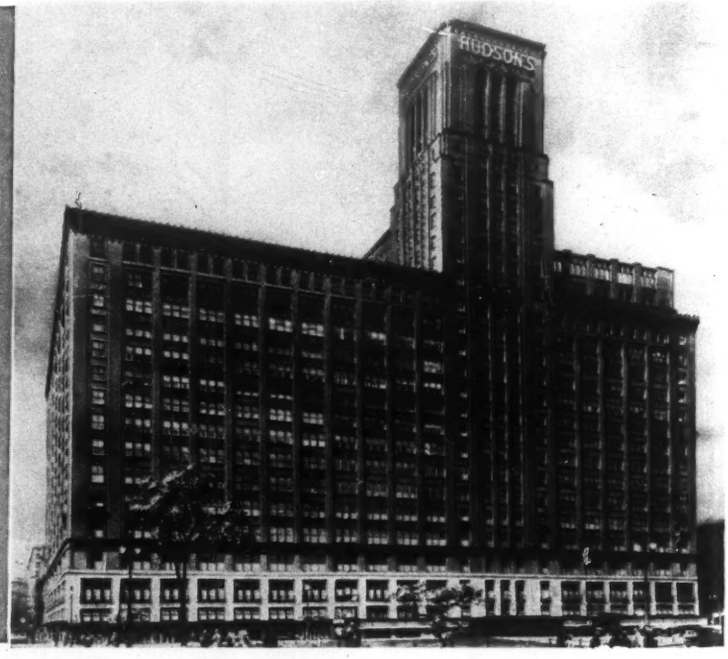
Before outlining the installations in the Hudson company buildings, the following pertinent facts may be of interest: The combined buildings occupy an entire rectangular city block and are 18 stories high, exclusive of the story heights of the tower. The original Farmer Street store has three basements, two of which are used as selling basements and the third for mechanical equipment. The Woodward Avenue store has two basements, and the newest building, known as the Newcomb-Endicott addition, is provided with four basements. Each floor above the street level comprises approximately 76,500 sq ft of sales area. The building is approximately 420 ft long by 220 ft wide, which would represent an area of 92,400; some of this area is of course taken up by elevators, service rooms and stair towers.

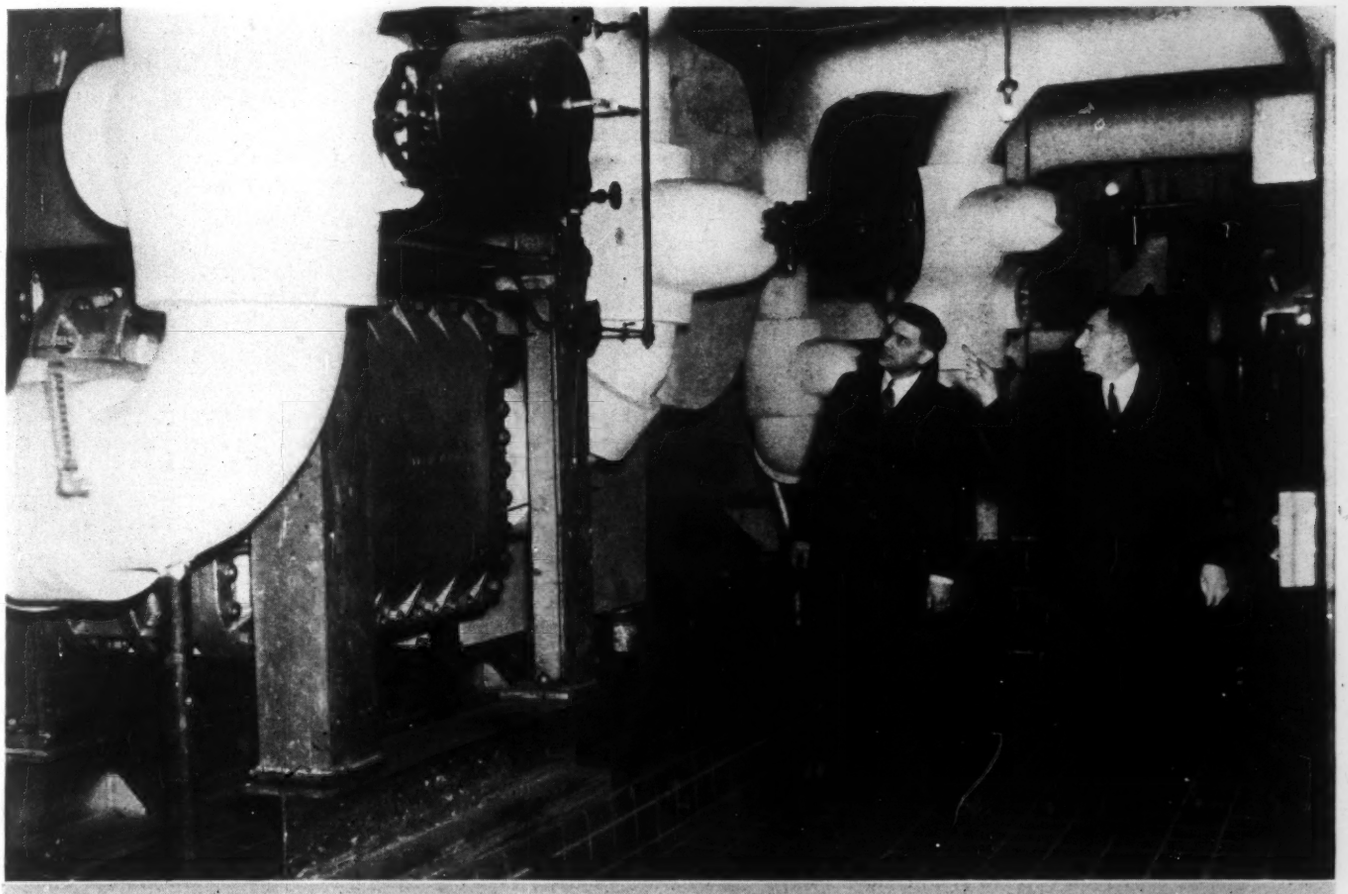
*Chief Mechanical Engineer, J. L. Hudson Co.
Illustrations courtesy Carrier Corp.

First Air Conditioning in 1925

The first installation for mechanically conditioned air—washed and cleaned, humidified or dehumidified, heated or cooled—was made in 1925. This comprises three centrifugal refrigeration machines in the third basement of the Farmer Street building, each having a capacity of approximately 250 tons per 24 hr, cooling water to 41 degrees. These machines are tied in with several central station air conditioning apparatuses (each consisting of fans, reheaters, preheaters, automatic oil filters and spray type air washing and dehumidifying equipment) that cool and condition the air. The air is distributed uniformly throughout the two basements, first floor and mezzanine by a system of duct work.

If anyone doubts the need for and importance of real engineering in the design, installation, operation and maintenance of an air conditioning system, this discussion by Mr. Warr will prove most illuminating. He describes the air conditioning of the J. L. Hudson department store in Detroit, a project which was initiated in 1925 when the first equipment was placed in operation and which has been extended since that time. . . . For instance, the design of the air conditioning for the fifth and sixth floor ready-to-wear departments was complicated by some 400 fitting and try-on rooms around the outside walls which isolate the interior sales areas from the influences of outside heat transmission and sun effect. . . . How the sun affects the air conditioning load of a building at various times of the day is indicated by Fig. 2; for the J. L. Hudson store, the maximum is at 5:00 p. m.





Building Superintendent Netting and Chief Mechanical Engineer Warr inspect some of the equipment

The second group of air conditioning apparatus was installed in 1928 at the time the Newcomb-Endicott addition was built. This equipment, in the third basement, provides comfort conditioning for the first floor and mezzanine of the Newcomb-Endicott building, the two basements and first floor of the Woodward Avenue building, and the three basements of the Newcomb-Endicott building which are used for sales purposes. This system comprises three centrifugal refrigeration machines, each having a capacity of approximately 350 tons, with air handling equipment for air cooling and air washing very similar to that used in the original Farmer Street basements, first floor and mezzanine areas.

At the time the Newcomb-Endicott building was constructed there was also installed on the 18th floor of this building one 200 ton centrifugal machine for air conditioning the public dining rooms on the 13th floor. The air conditioning and air handling equipment is of the open spray type and the spray water is cooled by the refrigeration machine to approximately 41 degrees.

All of these systems are of the conventional open spray type for year 'round air conditioning, in which the spray water is circulated from the dehumidifiers and air washer tanks by motor driven centrifugal pumps and delivered directly to the marine type coolers of the refrigeration machines. After the water has been chilled to the required temperatures, it is conveyed directly to the banks of spray nozzles in the dehumidifier chambers for conditioning the air drawn through the chambers by the fans.

During the cooling season the air is cooled and dehumidified to 50 degrees, corresponding to four grains of moisture per cubic foot. The conditioned areas are maintained at temperatures below 80 degrees and usually from 76 to 78 degrees which, with relative humidities ranging from 50 to 55 per cent provide the desired comfort conditions. During the heating season the water is recirculated without precooling or refrigeration and the air temperatures leaving the spray chambers are automatically regulated to hold approximately 40 degrees, thus humidifying the air and correcting for the moisture deficiency of the air taken from outdoors.

In some of the basement departments the heating effect from the lights is equivalent to 6 watts per sq ft of floor area. The light loads vary in the different departments; in some the light load is as low as 3 watts per sq ft.

The basements and first floor are not subject to the usual influences and variable air conditioning loads due to sun effect; nevertheless zoning of each of the departments was necessary in order to facilitate operation of the air conditioning system by varying the air supply temperatures to each department to meet the fluctuating load conditions due to people and occupancy. To meet these varying conditions the automatic control system fractionally regulates the position of the return and bypass system of air dampers during the summer months and, in addition, regulates by graduated action the steam admission to the reheating coils during the winter heating season. The volume of air supplied to each department is maintained practically constant the year 'round, but

the temperature of the air supplied is varied according to cooling or heating requirements.

The next major step taken by the Hudson company was the air conditioning of the fifth and sixth floor ready-to-wear departments, which presented many problems of a complex nature—due mainly to the fact that the fitting and try-on rooms, some 400 in number, are situated continuously all around and next to the outside walls of the building, completely isolating the inside and larger salons and sales areas from outside temperature and sun effect influences.

The sun effect and outside radiation influences present a variable and shifting load condition, affecting in the main the try-on and fitting rooms. This shifting and variable sun heat load similarly affects all of the other floors of the building, except that on other floors where there are no outside try-on and fitting rooms the sun effect is diffused to larger areas. Fig. 1 shows the position of the sun in respect to the building at various hours of the day, and is based on sun location and altitude as of July 25 in Latitude 40, eastern standard time.

Following intensive study on the part of our engineers and architects, it was clearly evident that a system capable of a wide range in flexibility together with coordinated zoning would be necessary to meet all the requirements.

The sun effect varies according to the season of the year and its intensity also varies according to sun altitude. This, when coupled with a shifting occupancy load, variables in outside temperatures that affect heat transmission rates, as well as infiltration factors, obviously demanded a most careful analysis and coordination of all factors before any definite conclusions could be reached as to the most practical design for air distribution, volume of conditioned air, zoning and arrangement and location of apparatus.

Maximum Sun Load at 5:00 P. M.

The maximum air conditioning load from sun gain alone occurs at about 5:00 p. m. and not at noon, as might be thought. This is due to the position of the building in relation to the position of the sun and to the fact that the sun heating effect is greatly intensified at this altitude against the broad side of our building, striking at almost right angles to the building wall and glass area. (See Fig. 2, which shows the load curve, sun gain only, at various hours of the day.)

Another interesting study made prior to the installation of our upper story air conditioning systems indicates the relation of the five principal factors that make up the total air conditioning load—transmission, infiltration, lights, people, and sun gain. These relations may vary considerably when analyzed against other department stores and office buildings, and Fig. 3 therefore pertains only to the Hudson store building.

It should be noted that all loads are pyramided and calculated on the basis of outside maximum conditions of 95 F dry bulb and 75 F wet bulb temperatures against inside design conditions of 80 F dry bulb and 50 per cent relative humidity with instantaneous maximum people and light load and variable sun effect load according to the time of day. The load curve for only transmission and infiltration, lights and people, would be substantially the same, less that indicated as sun gain according to the hours of the day. To illustrate, our maximum load may occur at 5 p. m., at which time the sun factor is equivalent to 22.4 per cent of the total load, people 19.4 per cent, lights 35.8 per cent, transmission and infiltration 22.4 per cent. These ratios vary according to the time of day.

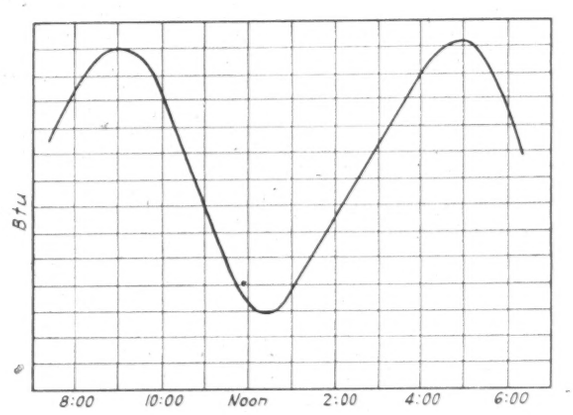


Fig. 2 — Load curve — sun gain only—at various hours of the day

Typical Scheme of Installing Equipment

To meet all of the requirements imposed, it was decided first to install the apparatus and equipment for the sixth floor, since this floor represented the most complex problem of any of our 15 sales floors, all of which are now conditioned. The arrangement shown in Fig. 4 is typical for all floors except the 10th, 11th and 12th.

The air conditioning equipment (consisting of zone fans, reheaters, cooling coils, air washing spray system,

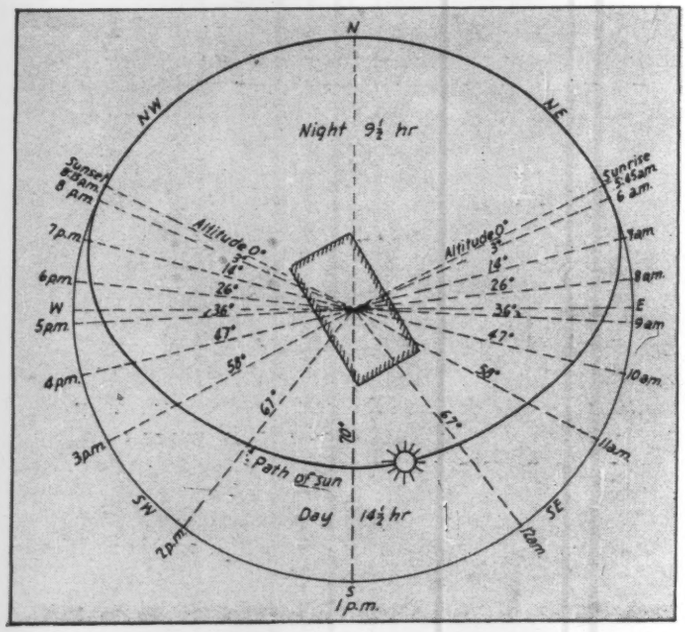


Fig. 1—Position of the sun in respect to building at various hours of the day

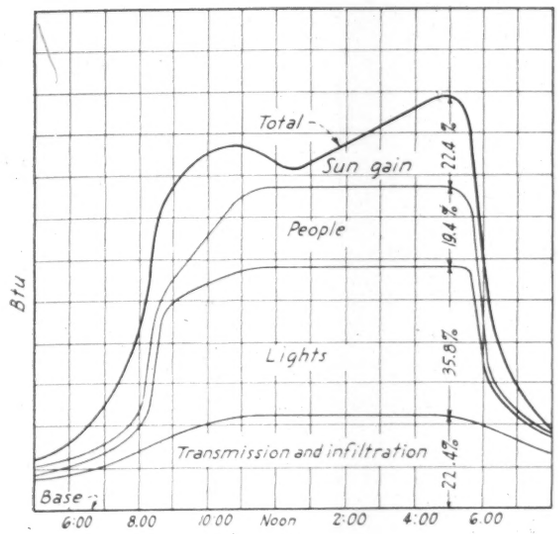


Fig. 3—Load curve — total of all factors—at various hours of the day

preheaters, air filters, automatically operated outside and return air dampers, motors and starters) is supported on concrete slab platforms erected at locations as shown on Fig. 4. These platforms are in no case less than 7 ft above the selling floor level, hence no valuable sales space is given up for the accommodation of equipment. Each apparatus room is fully enclosed and is provided with return air openings and sound absorbers.

The duct work extending from the various apparatus rooms for supplying conditioned air to the zones is fabricated of galvanized metal and encased in furring of expanded metal and plaster. The air outlets are of the high velocity directional flow type and are adjustable for air flow direction. Each zone—there being 14 on the sixth floor—is independently regulated by thermostatic control. The controls are actuated by the dry bulb temperature and regulate the position of the dehumidified air dampers and bypass dampers to vary the supply air temperatures to each zone according to the load variations. The volume of air supplied to all zones is fixed and remains constant.

The mechanical refrigeration to meet the entire air conditioning load requirements of the second to 12th floors, inclusive, is produced by four centrifugal refrigeration machines, two of which are 800 ton capacity each and two of which are 640 tons capacity each. These are situated in the third basement machinery room, adjacent to the three original machines which were installed in 1928 for the Newcomb-Endicott building addition. These machines are motor driven through speed increasing gears and are equipped with variable speed controllers. The last four machines, which total 2880 tons capacity, occupy a floor area of 76 ft by 18 ft, equivalent to 1368 sq ft of floor area.

The evaporators or water coolers are of the marine type; the water to be chilled passes through the admiralty metal tubes of the cooler while the refrigerant (Carrene) evaporation takes place around the tubes. The condensers are of the same general type, with the condensing water flowing through the tubes and the refrigerant condensing around them. City water, generally at 70 F, is used for condensing.

The refrigeration effect is transferred from the refrigerating machines to the various independent air cooling and dehumidifying apparatuses in the form of chilled water at 43 to 45 F through a closed water circulating system extending from the recirculating pumps in the fourth basement. Piping connections from the pump discharges deliver the water through the refrigerating machine coolers and from the coolers to the several 14 and 16 in. risers. Laterals from these risers are extended on each floor to the air treating equipment and the water, after passing through the air cooling coils, is returned in similar manner, through laterals and a return water piping system, to the pumps, as indicated in Fig. 5. Under full load conditions approximately 10,000 gpm of water is circulated by four centrifugal pumps operating against a head of 365 ft.

Bypass connections have been provided in the piping system with the necessary valves to permit the operation of all four machines simultaneously, or the operation of any combination of machines as may be demanded according to the load conditions. All chilled water piping is thoroughly insulated with ice water thickness cork.

Motors Controlled from Starting Board

With an air conditioning system of this magnitude, the matter of simultaneously starting and stopping the equipment, fan motors and pump motors, was worked out very simply. Since most of the individual fan

Fig. 4—Diagrammatic sketch showing typical method of air distribution and zoning

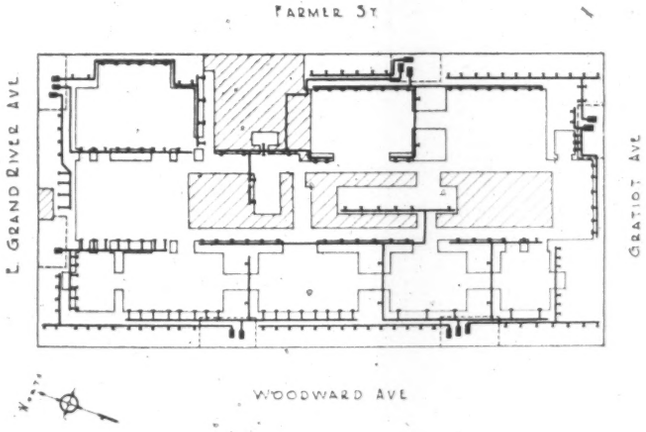
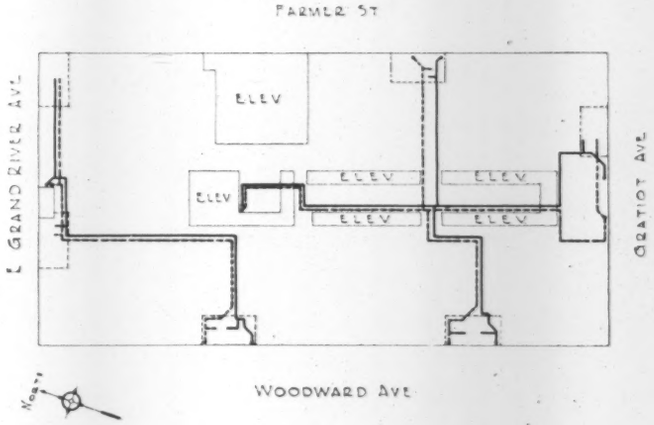


Fig. 5—Typical arrangement of chilled and return water piping



motors are of small horsepower, 7½ hp or less, an electric circuit is carried from each apparatus room to the third basement starting board, which is adjacent to the operating engineer's office. The circuits are cut in or out by remote pushbutton controls operating solenoid switches situated in each of the apparatus rooms. Thus, the individual fans and pumps are started. A telltale red light above each pushbutton indicates that the apparatus is in operation. Each motor can also be started or stopped from control switches in the apparatus room.

Proper Maintenance of Great Importance

Of great importance in obtaining good results with this large air conditioning plant, comprising many separate systems, are the following maintenance operations:

Thermostats (approximately 170) require to be cleaned and kept in good repair to assure correct floor temperatures.

Damper equipment which operates fractionally must be checked for proper movement and adjustments must be made.

Diaphragms for controlling steam pressure, air pressure, water quantities, dampers, heating coils, static pressure regulators, etc., must be kept in repair.

Filters of varied makes including oil, paper, spun glass, and throw-away types, require continuous cleaning, which work is set up in a regular schedule.

Blower fans require bearing inspections, repairs, and lubrication to assure remote control operation. Wheels and casings must be cleaned and painted at least once a year to prevent rust and corrosion.

Dehumidifiers call for careful attention, such as cleaning and flushing of tanks, cleaning spray nozzles, cooling coils, etc.

Pumps having capacities ranging from 100 to 3800 gpm call for mechanical attention, such as packings to carry 250 lb pressure, new shafts occasionally, new sealing rings to maintain proper clearances, etc.

Refrigerating machines ranging in capacities from 200 to 800 tons each are given annual general inspection to assure continuous uninterrupted operation. This involves removal of condenser and cooler heads, some of which weigh a ton, to allow tubes to be cleaned and checked for leaks.

Compressors having speeds of 5000 rpm driven through speed increaser gears by 250 to 1000 hp motors are inspected once a year for checking bearing wear, shaft seal repairs, cleaning lubricating system, and changing oil.

Air evacuating and purge systems are given yearly inspection to assure proper evacuation of air from the refrigeration system.

An interior—the beauty salon

Under full load conditions any of our refrigerating machines produces approximately 1 ton per hp. Water consumption for condensing ranges from 1 gpm per ton under light load conditions to 2 gpm under peak load conditions. All machines are electrically operated and involve over 150 motors ranging from ½ to 1000 hp in size.

After several years of experience in operating these systems, we feel that the results obtained have proved the value of our efforts and the careful analysis and designing of our architects and engineers to promote the comfort and health of our employees and customers.

Determining Relative Humidity at Low Temperatures

A READER asks about the method for determining relative humidity at temperatures below zero, and wants to know whether atomization or low pressure steam is the most effective method for humidifying a freezer at temperatures 25 degrees below zero.

REPLY BY E. VERNON HILL*—In determining the humidity below zero, the customary procedure is to use a whirling psychrometer. The wick covering the wet bulb is removed, the bulb itself wetted, and the water allowed to freeze, forming a thin coating of ice. The instrument is then whirled as in taking any wet and dry bulb reading, the evaporation of the ice giving the wet bulb depression.

At temperatures below freezing, the wet bulb depression is slight and must be read very carefully and the instrument swung a sufficient length of time to be sure the mercury has descended to the lowest point.

Hair hygrometers may be used to determine humidities below freezing, but are inaccurate unless carefully designed and frequently calibrated against a wet and dry bulb instrument.

With regard to humidification at low temperature, either atomizing or low pressure steam would be satisfactory. Of course with low pressure steam the heat from the steam would have to be taken into consideration in maintaining the proper temperature of the air.

There should be no difficulty in getting a high humidity at these low temperatures as the moisture content of saturated air at 25 F below zero is very low; it is only 0.47 grains per cu ft at zero.

*Member of Board of Consulting and Contributing Editors.

New Data for

Refrigeration Piping

Ammonia Suction Line Piping

By R. C. Doremus*

Basic data on the design and installation of piping for industrial and commercial refrigeration are presented in this series of articles. As a refrigeration engineer, Mr. Doremus has had the opportunity to observe jobs which have been poorly designed, has concluded that lack of information is the cause. He therefore makes his data available in this form

AMMONIA suction lines, like those for other refrigerants, should be designed and installed on the basis of pressure drop between the evaporators and compressor. In small plants, a pressure drop of from 2 to 4 or 5 lb is inconsequential for there is neither much tonnage nor horsepower involved; in larger plants, a pressure drop of 5 lb would be a considerable item of inefficiency for it would represent a large loss in tonnage capacity or of horsepower expended. Therefore, an ammonia suction line in a large plant (100 tons or more) should be based on a maximum total suction line pressure drop of 2 to 2½ lb.

Whether the ammonia compressor is vertical or horizontal, the fact that it is of a reciprocating type means that the peak velocity of gas passing from the suction line through the suction valve into the cylinder will be 1½ to 2½ times the suction line velocity, but with modern plate valve design, this step up in velocity causes a pressure drop of only ¼ to ½ lb, according to compressor speed. Thus, if the above line pressure drop is used as a gage and the suction strainer is reasonably clean, the total pressure drop from the inside of the evaporator to the inside of the compressor cylinder will not exceed 3 lb.

In many cases, the engineer or contractor knows very little regarding pressure drop of ammonia vapor in a pipe line. It has never bothered him, so he follows the line of least resistance and lets well enough alone. He may guess at the proper size suction line to be used for an installation or follow precedent of some previous job (if he has one to refer to) but in doing so, he is more of a layout man than a refrigerating engineer. He may guess right if he is a good guesser but is more apt to be wrong, with sad consequences. If the suction line he chooses is insufficient in cross sectional area, the gas velocity will be too high and the pressure drop between the evaporator and compressor will be still higher, for the pressure drop increases with the square of the velocity.

Lacking experience as a basis for design, he may simply equalize the suction pipe connection size on the compressor if there is only one, or total them up if there are several. It should be remembered, however, that the compressor builder is a practical manufacturer of machinery and does not know what field conditions will be encountered. He must build a machine for John Doe to operate at −40 F where the machine will have to handle a displacement 19,020 cu in. per minute per ton at a very high compression ratio and the next compressor on his assembly line must be shipped to Joe Jones where it will operate at +40 F handling a displacement of only 2875 cu in. at a very low compression ratio. One job may have a long suction line, irregular in shape, covering several floors of a multistory plant while the other may have a very short run on one floor. Obviously the pipe line that will be satisfactory for the one case will be inadequate for the other.

Components of Pressure Drop

In any ammonia suction line, the total pressure drop from evaporator to compressor will be the sum of the component pressure drops through pipe, valves and fittings, and it is, therefore, essential to know what these components are.

Table 1 shows the pressure drop of saturated ammonia vapor due to passing through elbows, tees and valves in which the amount of this drop is expressed as an equivalent length of pipe of the same size. This will enable anyone considering a certain layout of equipment or piping design to count the valves and fittings, equate them to an equivalent length of straight pipe, add this to the actual length of straight pipe and obtain the total length of pipe represented in the ammonia suction line.

Figs. 1 and 2 are charts showing the pressure drop of saturated ammonia vapor through 100 ft of pipe of various sizes according to the tonnage of refrigeration capacity carried at 0 F. Figs. 3 and 4 are similar charts showing the pressure drop of saturated ammonia vapor for a higher suction pressure equivalent to plus 41 F. It probably should be emphasized that none of these values is guaranteed to be absolutely accurate, for they

Table 1—Pressure Drop of Ammonia Suction Vapor Due to Valves and Fittings Expressed in Equivalent Length in Feet of Straight Pipe of Same Size

Pipe Size, In.	⅜	½	¾	1	1¼	1½	2	2½	3	3½	4	5	6	8	10	12
Elbow	1	1	2	2	2	2	3	4	5	6	7	8	10	15	18	20
Tee	1	1	2	2	3	3	4	5	6	7	8	10	12	20	25	30
Valve	2	2	3	3	4	4	5	6	8	9	10	15	20	25	30	45

*Chief Engineer, Detroit Ice Machine Co. Member of Board of Consulting and Contributing Editors.

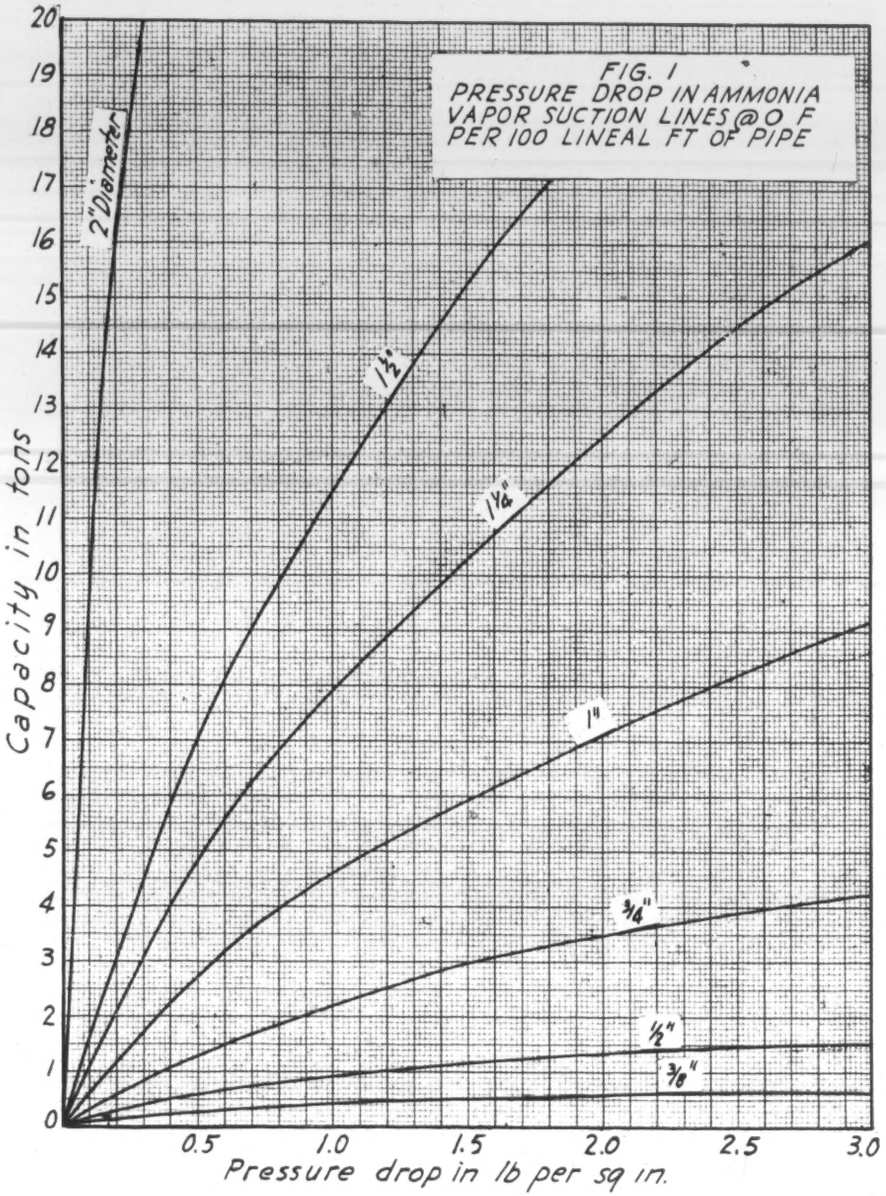


FIG. 1
PRESSURE DROP IN AMMONIA VAPOR SUCTION LINES @ 0 F PER 100 LINEAL FT OF PIPE
Capacity in tons
Pressure drop in lb per sq in.
2" Diameter
1½"
1¼"
1"
¾"
½"
⅜"
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
17
18
19
20
0.5
1.0
1.5
2.0
2.5
3.0

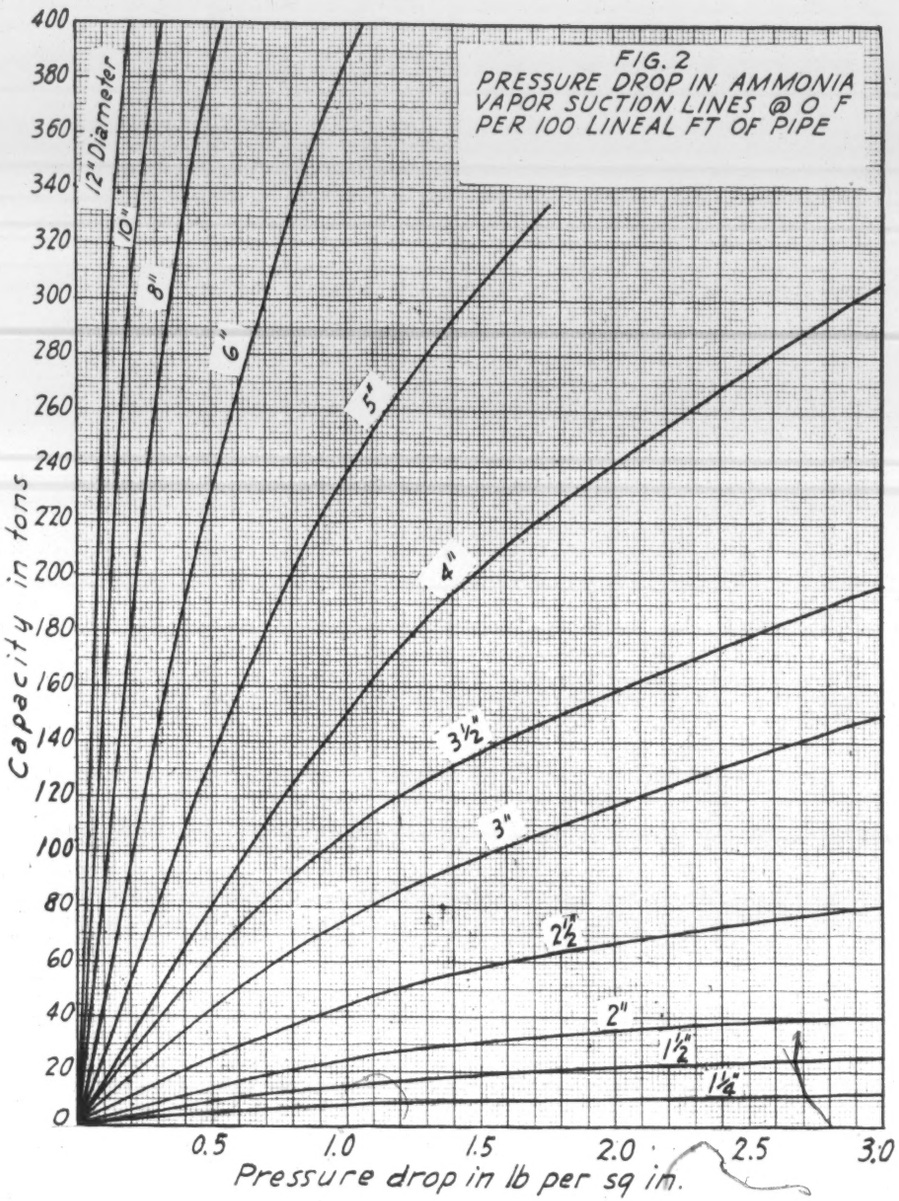


FIG. 2
PRESSURE DROP IN AMMONIA VAPOR SUCTION LINES @ 0 F PER 100 LINEAL FT OF PIPE
Capacity in tons
Pressure drop in lb per sq in.
12" Diameter
10"
8"
6"
5"
4"
3½"
3"
2½"
2"
1½"
1¼"
0
20
40
60
80
100
120
140
160
180
200
220
240
260
280
300
320
340
360
380
400
0.5
1.0
1.5
2.0
2.5
3.0

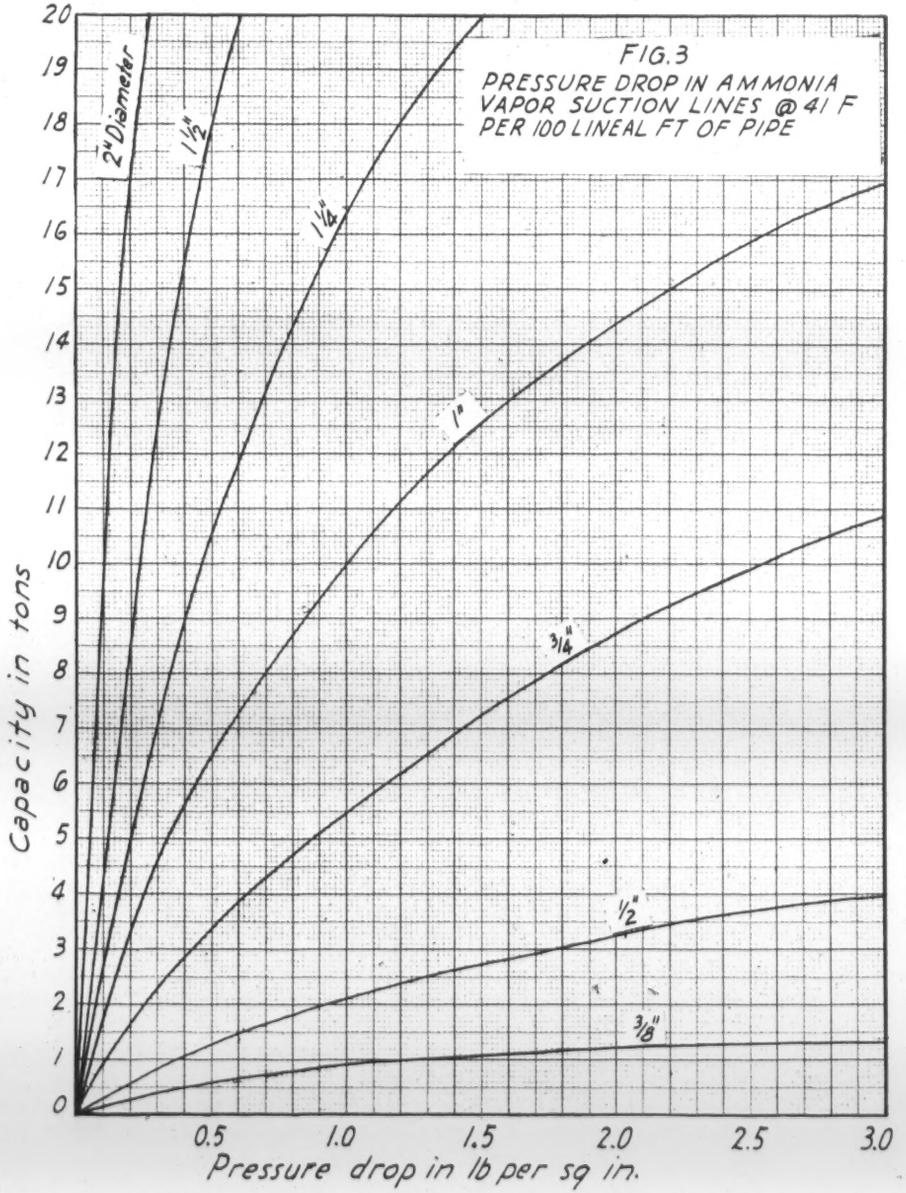


FIG. 3
PRESSURE DROP IN AMMONIA VAPOR SUCTION LINES @ 41 F PER 100 LINEAL FT OF PIPE
2" Diameter
1½"
1¼"
1"
¾"
½"
⅜"
Capacity in tons
Pressure drop in lb per sq in.
0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20
0.5 1.0 1.5 2.0 2.5 3.0

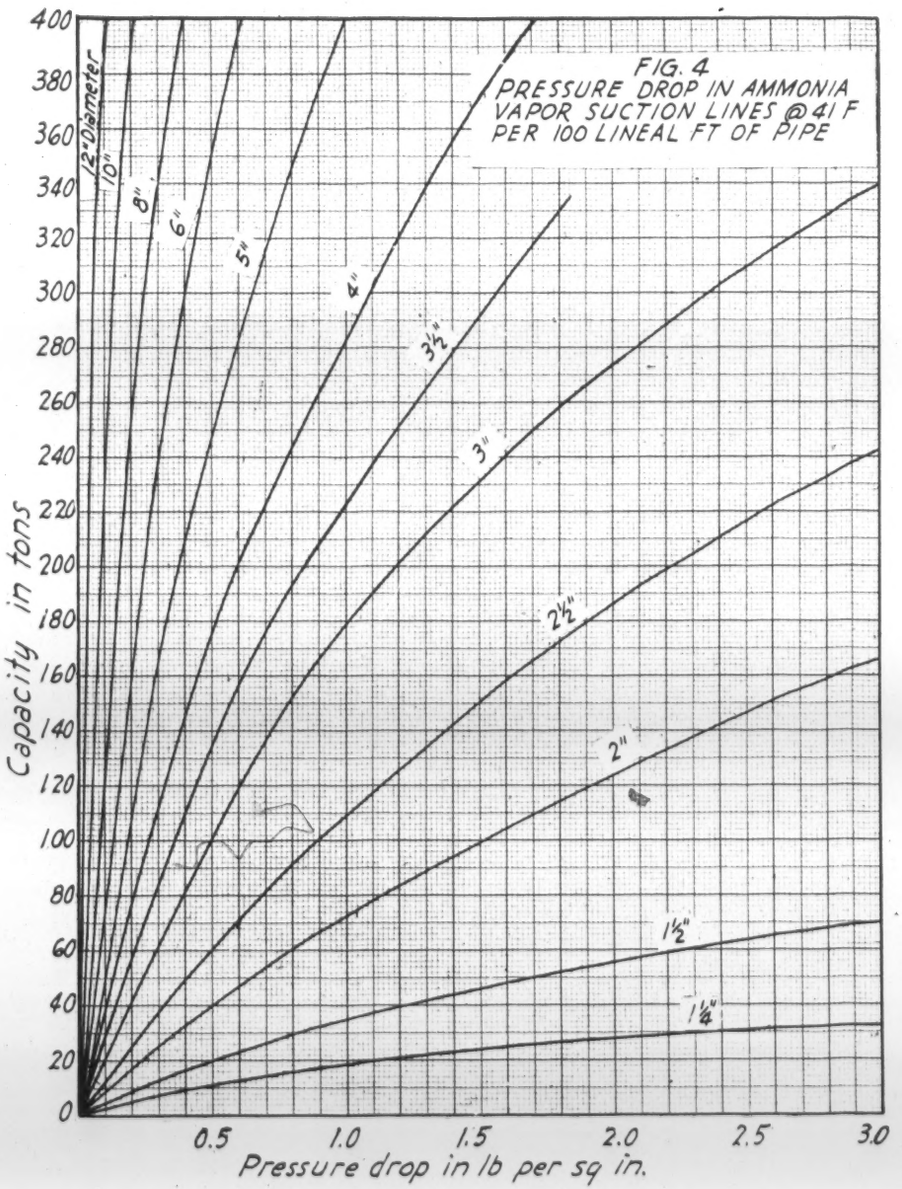


FIG. 4
PRESSURE DROP IN AMMONIA VAPOR SUCTION LINES @ 41 F PER 100 LINEAL FT OF PIPE
12" Diameter
10"
8"
6"
5"
4"
3½"
3"
2½"
2"
1½"
1¼"
Capacity in tons
Pressure drop in lb per sq in.
0 20 40 60 80 100 120 140 160 180 200 220 240 260 280 300 320 340 360 380 400
0.5 1.0 1.5 2.0 2.5 3.0

Table 2—Velocity Chart for Ammonia Suction Vapor at Various Suction Temperatures for Steel Pipe of Different Diameters, in Feet Per Minute Per Ton of Refrigeration

Temp. Saturated F	Corr. Gal. Press. Lb per Sq In.	Sp. Vol. of Sat. Vapor Cu Ft per Lb	Heat of Sat. Vapor Btu per Lb	Heat of Liq. @ 92 F Btu per Lb	Refrig. Effect Btu per Lb	Weight Lb per Min per Ton	Actual Displacement Cu In. per Min per Ton	Nominal Pipe Sizes, In.												
								Extra Heavy Pipe						Full Weight Pipe						
								3/8	1/2	3/4	1	1¼	1½	2	2½	3	3½	4	5	6	8	10
—40	8.75*	24.86	597.6	145.8	451.8	4425	19020	11270	6790	3665	2207	1235	900	472	331	215	160	124	79.2	55.0	31.6	20.1
—35	5.38*	21.68	599.5	145.8	453.7	4420	16520	9780	5890	3180	1916	1073	780	411	287	186	139	108	68.9	47.7	27.4	17.5
—30	1.63*	18.97	601.4	145.8	455.6	4390	14390	8520	5125	2760	1668	935	680	356	250	162	121	94.2	60.0	41.5	23.8	15.2
—25	1.3	16.66	603.2	145.8	457.4	4373	12620	7470	4500	2428	1463	820	595	313	220	142	106	82.6	52.6	36.4	20.9	13.4
—20	3.6	14.68	605.0	145.8	459.2	4356	11070	6550	3930	2128	1282	718	523	274	192	125	93.3	72.5	46.1	32.0	18.3	11.7
—15	6.2	12.97	606.7	145.8	460.9	4345	9710	5725	3460	1868	1128	631	458	242	169	109	82.4	63.7	40.5	28.0	16.1	10.3
—10	9.4	11.50	608.5	145.8	462.7	4323	8580	5060	3050	1648	995	557	404	213	149	96.2	73.2	56.2	35.7	24.7	14.2	9.1
—5	12.22	10.23	610.1	145.8	464.3	4314	7620	4490	2710	1463	885	495	359	189	132	86.4	64.8	50.0	31.7	22.0	12.6	8.1
0	15.7	9.12	611.8	145.8	466.0	4296	6760	3980	2400	1300	785	434	318	168	118	76.3	56.4	44.3	28.2	19.5	11.2	7.1
5	19.7	8.15	613.3	145.8	467.5	4276	6020	3550	2150	1158	700	392	284	150	105	68.0	51.1	39.4	25.1	17.4	10.0	6.4
10	23.8	7.30	614.9	145.8	469.1	4266	5390	3170	1920	1035	625	350	254	134	94.0	60.6	45.0	35.3	22.4	15.5	8.9	5.7
15	28.4	6.56	616.3	145.8	470.5	4252	4840	2850	1725	930	562	314	228	120	84.1	54.5	41.0	31.6	20.1	14.0	8.0	5.1
20	33.5	5.91	617.8	145.8	472.0	4240	4335	2550	1540	872	502	282	204	108	75.5	49.0	36.5	28.4	18.0	12.6	7.2	4.6
25	39.0	5.33	619.1	145.8	473.3	4227	3895	2295	1388	748	452	253	183	96.8	67.8	43.8	32.7	25.5	16.2	11.2	6.4	4.1
30	45.0	4.83	620.5	145.8	474.7	4218	3510	2070	1250	674	407	228	166	87.1	61.0	39.5	29.5	23.0	14.6	10.2	5.8	3.7
35	51.5	4.37	621.7	145.8	475.9	4205	3175	1870	1130	610	368	206	150	79.0	55.3	35.7	26.7	20.8	13.3	9.2	5.3	3.4
40	58.6	3.97	623.0	145.8	477.2	4190	2875	1695	1025	552	334	187	136	71.5	50.0	32.4	24.3	18.8	12.0	8.3	4.8	3.1
41	60.1	3.90	623.2	145.8	477.4	4188	2820	1660	1000	542	327	183	133	70.0	49.1	31.7	23.7	18.5	11.7	8.1	4.7	3.0
Maximum velocity recommended for tonnage capacity in any pipe size to avoid excessive pressure drop in suction mains								1000	1400	2000	2500	2750	3000	3300	3600	4000	4200	4400	4700	5000	5400	6000

*Vacuum in inches.

Note: All pipe sizes 1½ in. and smaller are based on extra heavy weight steel pipe.
All pipe sizes 2 in. and larger are based on full weight steel pipe.

Example: What size ammonia suction line should be used for a load of 140 tons at a temperature of —5 F?
If we consider a 4 in. pipe, 140 x 50 = 7000 fpm.
This is too high a velocity; limit for 4 in. is 4400 fpm.
If we consider a 5 in. pipe, 140 x 31.7 = 4438 fpm.
This 5 in. is suitable inasmuch as its limit is 4700 fpm.

are simply gathered from the best sources of present available information. Some of the data are from laboratory tests, some from field observations and those portions on which little or no information was available were interpolated or extrapolated to complete the curves.

To illustrate the use of these data, assume we are considering an ice manufacturing plant in which there is a load of 200 tons of refrigeration with a suction vapor pressure of 15.7 lb per sq in. gage, and the suction line is 225 ft long and includes 8 elbows, 6 tees and 3 valves.

If we assume a 5 in. suction line to be satisfactory and check it from the tables and curves:

8 elbows @ 8 ft	= 64 ft in equivalent straight pipe
6 tees @ 10 ft	= 60 ft in equivalent straight pipe
3 valves @ 15 ft	= 45 ft in equivalent straight pipe
Suction pipe line	= 225 ft in equivalent straight pipe
Total	= 394 ft in equivalent straight pipe

From the curves, 200 tons capacity in a 5 in. suction line represents a drop of 0.80 lb per 100 ft of length. The total pressure drop in this case will be:

$(394 \div 100) \times 0.80 = 3.15$ lb, which is too high.

If we check it in a 6 in. suction line, we find:

8 elbows @ 10 ft	= 80 ft in equivalent straight pipe
6 tees @ 12 ft	= 72 ft in equivalent straight pipe
3 valves @ 20 ft	= 60 ft in equivalent straight pipe
Suction pipe line	= 225 ft in equivalent straight pipe
Total	= 437 ft in equivalent straight pipe

From the curves of pressure drop, 200 tons of refrigeration in a 6 in. ammonia suction line at 15.7 lb or 0 F saturated has a drop of 0.42 lb per 100 ft of length. The total pressure drop in this case will be:

$(437 \div 100) \times 0.42 = 1.84$ lb, which is satisfactory.

Therefore, 6 in. should be used for this suction line. If 5 in. is used, it is not much too small, but the pressure drop is higher than approved in standard practice and the compressor capacity will be diminished about 3 per cent and the power cost per ton of refrigeration will be about 4 per cent too high. These percentages would not mean much in a smaller plant, but they represent considerable items in this size plant and should be properly considered before the piping is specified or installed.

However, suppose we had this same 200 ton load for air conditioning in a rayon plant, operating at an evaporator temperature of 41 F with a suction line only 100 ft long containing 3 elbows, 1 tee and 1 valve. Let us check this for a 4 in. suction pipe:

3 elbows @ 7 ft	= 21 ft in equivalent straight pipe
1 tee @ 8 ft	= 8 ft in equivalent straight pipe
1 valve @ 10 ft	= 10 ft in equivalent straight pipe
Suction pipe line	= 100 ft in equivalent straight pipe
Total	= 139 ft in equivalent straight pipe

The pressure drop of a 200 ton load in a 4 in. suction pipe line operating at 41 F saturated is 0.60 lb per 100 ft length. The total pressure drop will then be:

$(139 \div 100) \times 0.60 = 0.83$ lb per sq in. gage, which is satisfactory.

This shows that in the one case, a 6 in. ammonia suction pipe line is required for good design, while in another plant, a 4 in. suction line will be satisfactory. Conversely, while the 4 in. suction line is satisfactory for the second

case, it would be utterly inadequate for the first case as shown below:

8 elbows	@ 7 ft	= 56 ft in equivalent straight pipe
6 tees	@ 8 ft	= 48 ft in equivalent straight pipe
3 valves	@ 10 ft	= 30 ft in equivalent straight pipe
Suction pipe line		= 225 ft in equivalent straight pipe
Total		= 359 ft in equivalent straight pipe

(359 ÷ 100) × 1.47 = 5.28 lb pressure drop, which is much too high.

Capacity of Suction Line Piping

Figs. 1, 2, 3 and 4 show pressure drops for various tonnage capacities in different size pipe lines at only two suction pressures, whereas there are many back pressures that might be charted to suit all conditions. However, for practical use, these may be interpolated.

It follows that there is a maximum capacity in tons of refrigeration that may be carried in any size pipe to give a reasonably low pressure drop, and inasmuch as a pipe of any size has a definite cross sectional area, it resolves itself into a question of maximum gas velocity for each size pipe. Table 2 is presented to show the actual displacement in cubic inches of saturated ammonia required per minute per ton at different back pressures corresponding to a range from minus 40 F to plus 41 F, together with the velocity of the suction vapor in feet per minute per ton; this range will cover the ordinary case encountered.

It will be observed that there is quite a variation in the displacement of suction vapor required in cubic inches per minute per ton of refrigeration and this is principally caused by the variation in the specific volume of the vapor in cubic inches per pound of ammonia at these different evaporator pressures. This is naturally reflected in the same proportion in the vapor velocity in feet per minute per ton of refrigeration as tabulated under each size pipe. Near the bottom of the table, there is given the author's notes on the maximum recommended velocity for any size pipe. Thus, if the maximum is divided by the velocity per ton tabulated, we can determine the capacity in tons that any size pipe will accommodate under various suction pressure conditions.

One advantage in this type of table is that it is prepared on a unit basis per ton so that the engineer may determine velocities quickly for any capacity, avoiding the necessity of making longer calculations and the possibility of error. Also, if he chooses to permit a higher maximum velocity than recommended, he can very easily do so. These recommended maximum vapor velocities are based on an average plant with a reasonably short suction line installed in a direct manner with a minimum of fittings, valves or offsets and the table is therefore intended for quick reference on this type of job. If the plant under consideration has an extremely short direct suction pipe line, higher tonnage capacities could be used without detriment. But if the suction line is unusually long and includes very many fittings, valves, etc., the pressure drop should be carefully checked by Table 1 and Figs. 1, 2, 3 and 4 before a size is definitely specified.

For example, if we are considering a 100 ton installation operating at a suction pressure of 15.7 lb which is 0 F saturated and the suction pipe line is reasonable short and direct, what size should the suction line be?

Referring to Table 2 we find:

100 × 44.3 = 4430 fpm. This velocity is satisfactory for a 4 in. line.

However, suppose we had the same 100 tons capacity in another plant operating at 8.75 in. vacuum, or —40 F saturated:

100 × 124 = 12400 fpm. This velocity is far too high for a 4 in. pipe.

100 × 79.2 = 7920 fpm. This velocity is even too high for a 5 in. pipe.

100 × 55.0 = 5500 fpm. This velocity is even too high for a 6 in. pipe, although it should be checked for, it is close to the limit.

100 × 31.6 = 3160 fpm. This is below 5400 fpm and an 8 in. pipe is satisfactory without any question.

Now, suppose we had the same 100 ton load in still another plant doing air conditioning at a pressure of 60.1 lb gage, or 41 F saturated:

100 × 31.7 = 3170 fpm. This is satisfactory for a 3 in. pipe.

Thus, the same 100 ton load at one condition requires a 3 in. suction pipe, in another case it requires a 4 in. suction line, and in still another case either 6 or 8 in. This indicates the importance of carefully checking each installation.

Suppose we have three evaporators, each with a short branch suction line joined in one suction main line, each operating at 15.7 lb pressure, or 0 F saturated. The loads are 18, 95 and 95 tons.

For the first evaporator—18 ton capacity:

18 × 168 = 3024 fpm. This is satisfactory for a 2 in. branch suction line.

For the 2nd evaporator—95 ton capacity:

95 × 44.3 = 4208 fpm. This is satisfactory for a 4 in. branch suction line.

Joining these two in one suction line:

113 × 44.3 = 5006 fpm. This is too high for a 4 in. suction line.

113 × 28.2 = 3187 fpm. This is satisfactory for a 5 in. suction line.

For the 3rd evaporator—95 ton capacity:

95 × 44.3 = 4208 fpm. This is satisfactory for a 4 in. suction line, as above.

Joining this to a 5 in. suction line:

208 × 19.5 = 4056 fpm. This is satisfactory for a 6 in. suction line.

Therefore, we should use a 6 in. suction line at the compressor graduated in size to 5, 4 and 2 in. in order to maintain a suitable gas velocity and friction drop, regardless of the suction pipe size furnished on the compressor.

The value of such a chart of gas velocities is clearly evident, but it should be emphasized that its principal use is in making design approximations and that each case—especially those that are close to the limit, or doubtful—should be rechecked by Figs. 1, 2, 3 and 4.

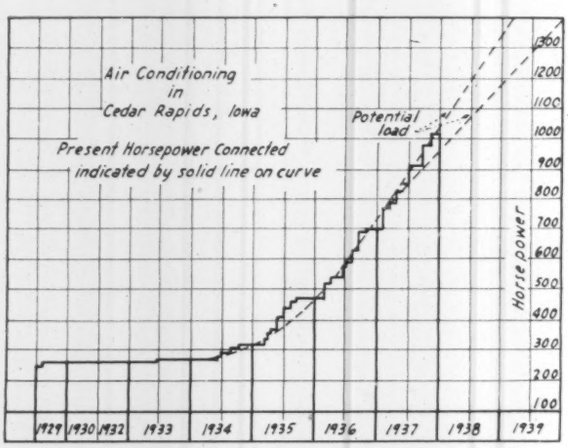
For data on piping other refrigerants, see the following issues of Heating, Piping and Air Conditioning: Ammonia Liquid Line Piping, December, 1937; "Freon" Liquid Line Piping, August, 1937; "Freon" Suction Line Piping, September, 1937; Methyl Chloride Liquid Line Piping, October, 1937; Methyl Chloride Suction Line Piping, November, 1937.

Heating • Piping and Air Conditioning's Annual Air Conditioning Review

1937 Installations 'Way Ahead of '36

IT is hardly news that 1937 air conditioning installations exceeded 1936 jobs by wide margins, as this has been evident from the preliminary reports issued by the utilities and manufacturers' groups during the year. The table on the next page, which follows the form of surveys for previous years published in HEATING, PIPING AND AIR CONDITIONING, shows just what the gains have been in various areas. The figures given were obtained from the electric utility companies, whose courtesy in making them available for publication is gratefully acknowledged.

Comprehensive data of this nature are still difficult to compile because the information is not always available, and because there is no one form or method universally used. Records are naturally more complete for air conditioning installations supplied with energy by the utility, and for this reason it is probable that there are many jobs—particularly industrial—which are not included. Also, where installations have other than motor driven refrigeration machines (such as steam jet, well water, adsorption, etc.) the connected horsepower figures do not give an adequate picture of the air conditioning capacity; therefore, an attempt has been made, where possible, to amplify the tabulated figures in the footnotes. However, the data for the different years should be strictly comparable for any one city, as they were obtained from the same source.



How air conditioning is growing in one city

Some information on gas using air conditioning systems may be had from the process and comfort air conditioning committee report of the American Gas Association. Lists of installations of gas fueled engines in air conditioning service, and gas dehumidifier jobs, are given.

The Knickerbocker Ice Co., serving air conditioning installations in New York, Philadelphia, Boston, Baltimore, and Washington, reports that there are now operating in its territories 85 ice cooled air conditioning systems, and that 35 per cent of this equipment was installed during 1937. These 85 installations represent a total refrigeration capacity of 4000 tons (*i.e.*, 4000 tons of compressor capacity would be required if they were compressor rather than ice using jobs), and the connected horsepower for the fan and circulating water pump motors on these installations probably amounts to about 800 hp.

As to railroad air conditioning, the Association of American Railroads advises that on October 1, 1937, the Class I railroads had 5196 and the Pullman company had 4666 air conditioned passenger cars in operation. These figures include sleeping cars, lounge cars, and other passenger equipment. In the first nine months of 1937, air conditioning equipment was installed on 1784 passenger cars, of which 1270 were railroad owned and 514 Pullman owned. Two hundred and fifty-six new passenger cars were on order for Class I railroads and 171 cars for the Pullman company on October 1 of this year, practically all to be air conditioned and completed by the end of 1937.

As of March 1, 1937, the number of air conditioned cars owned by 55 railroads in the United States and Canada was as follows:

Electro-mechanical	1042
Direct mechanical	537
Internal combustion	71
Head end power articulated trains	54
Steam ejector	809
Ice activated	1747
Total	4260

The Outlook for '38

Predictions are "dangerous" but interesting and a number of attempts have been made to compile estimates of what air conditioning will do in 1938. For instance, *Air Conditioning Trends*, issued by the publisher of HEATING, PIPING AND AIR CONDITIONING, recently asked ten sales executives in the air conditioning field to give their opinions of the five markets that will be most

Number of Air Conditioning Installations and Connected Horsepower for Air Conditioning for Commercial and Institutional Buildings, Industrial Plants, and Residences for 1936, 1937, and Totals as of Dec. 1, 1937

Area	1936								1937 (11 Months)								Totals as of Dec. 1, 1937								Approx. Population of Area
	All Bldgs.		Com. & Inst.		Industrial		Residential		All Bldgs.		Com. & Inst.		Industrial		Residential		All Bldgs.		Com. & Inst.		Industrial		Residential		
	No.	Hp	No.	Hp	No.	Hp	No.	Hp	No.	Hp	No.	Hp	No.	Hp	No.	Hp	No.	Hp	No.	Hp	No.	Hp	No.	Hp	
ALABAMA																									
Birmingham	44	594	*	*	*	*	*	*	61	1652	59	1650.5			2	1.5	151	4693	*	*	*	*	*	*	350,000
CONNECTICUT																									
Hartford	15	360	14	357			1	3	27	275	26	274.25			1	0.75	70	1656.5	63	1514.75	3	135	4	6.75	247,000
DIST. OF COLUMBIA																									
Washington	168	10251.25	129	10214			39	37.25	174	5808	160	5719.5	1	68	13	20.5	678	45250.25	557	44832.25	9	224	112	194	627,000
FLORIDA																									
Fla. P. & L. Co. System	84	994	*	*	*	*	*	*	95	1016	*	*	*	*	*	*	256	3935	*	*	*	*	*	*	400,000
St. Petersburg	2	10	2	10					16	351	16	351					22	381	22	381					50,000
ILLINOIS																									
Chicago	433	8734	351	7945	34	747	48	42	562	11610	488	10339	17	1184	57	87	1918	69237	1694	67052	51	1931	173	254	3,500,000
Rockford	*	80.5	*	*	*	*	*	*	*	287.75	*	*	*	*	*	*	*	*	*	*	*	*	*	*	85,900
INDIANA																									
Indianapolis and Marion Co.	79	874.2	49	798.95	1	35	29	40.25	113	1259.2	87	1213.7	3	22	23	23.5	280	5994.7	203	5346.2	9	534.5	68	114	400,000
IOWA																									
Cedar Rapids	21	201.5	18	194			3	7.5	22	304.5	20	300			2	4.5	69	1019.5	53	963.25	2	24	14	32.25	56,000
Des Moines	69	2245.76	51	2223.43	1	1	17	21.33	95	1773.56	74	1748.33			21	25.23	258	5325.42	202	5201.36	3	38.5	53	85.56	150,000
KENTUCKY																									
Louisville	43	530.16	*	*	*	*	*	*	89	1495.54	65	1210.62	8	251.8	16	33.12	217	4059.7	*	*	*	*	*	*	350,000
MARYLAND																									
Baltimore	75	1595.5	63	1549	2	9.5	10	37	199	3435	132	3232	4	131.5	63	71.5	493	13001.5	387	12111	17	745	89	145.5	1,145,000
MASSACHUSETTS																									
Boston Edison Co.	93	2243	*	*	*	*	*	*	243	1530	*	*	*	*	*	*	501	12556	*	*	*	*	*	*	1,394,000
MICHIGAN																									
Detroit & surroundings	236	4249	124	4066.7	6	68.8	106	113.5	329	4104.8	198	3861.5	7	104.5	124	138.8	834	24593.7	521	21795	35	2439.6	278	359.1	2,460,000
Lower, except Detroit	143	1365	126	1350			17	15	118	1931	94	1483	16	430	8	18	261	3296	220	2833	16	430	25	33	950,000
MINNESOTA																									
Minneapolis	*	*	*	*	*	*	*	*	137	1946	121	1917	1	3	15	26	425	4469	312	4303	7	23	106	143	465,000
MISSOURI																									
Kansas City	234	2206.55	133	2014.88			101	191.67	254	4757.31	153	4142.63	4	418.25	97	196.43	836	14832	498	13626.82	7	489.25	331	715.93	425,000
St. Louis	449	7862.17	242	7166.23	10	386.65	197	309.29	588	11046.29	315	9861.76	18	650.28	255	534.25	1545	33067.51	836	29084.54	61	2707.72	648	1275.25	1,250,000
NEBRASKA																									
Omaha	88	1300	46	1230			42	70	117	1587	92	1504			25	83	297	5306	180	4432	22	651	95	223	220,000
NEW HAMPSHIRE																									
Manchester	4	46.75	3	24.75	1	22			4	35.5	3	34			1	1.5	16	140.95	11	112.7	4	26.75	1	1.5	153,000
NEW YORK																									
Ithaca district	2	22.75	1	22			1	0.75	8	50.5	7	49.75			1	0.75	11	81.75	9	80.25			2	1.5	25,000
New York City	*	*	*	*	*	*	*	*	415	16203.5	*	*	*	*	*	*	*	*	*	*	*	*	*	*	6,772,000
Rochester	40	1702.75	24	415.25	6	1270	10	17.5	51	2406	28	425.5	14	1966.25	9	14.25	140	29187.25	81	3545	29	25589.25	30	53	450,000
Syracuse	41	1569.58	35	1345.33	5	222.5	1	1.75	22	233.7	18	216.7	1	8	3	9	63	1803.28	53	1562.03	6	230.5	4	10.75	218,000
OHIO																									
Cincinnati	172	1693	102	1592.25	*	*	70	100.75	188	2963.5	112	2861.25	*	*	76	102.25	605	12642.75	388	12330	*	*	217	312.75	750,000
Cleveland	*	*	*	*	*	*	*	*	19	16487.5	*	*	*	*	*	*	*	*	*	*	*	*	*	*	900,000
Dayton	53	750.96	25	338.03	14	396.93	14	16	73	1131.15	35	904.34	16	192.99	22	33.82	189	3545.21	95	2514.9	43	944.01	51	86.3	350,000
Toledo	48	1213	41	1087	3	120	4	6	41	390	28	220	7	165	6	5	140	3566	105	2961	15	577	20	28	300,000
OKLAHOMA																									
Okla. G. & E. System	160	1530	90	1412	1	4	69	114	193	2037	98	1835	5	24	90	178	484	6004	255	5541	9	45	220	418	590,800
P.S.Co. of Okla. System	33	393	10	321			23	72	101	6813	39	6407	4	278	58	128	203	9241.5	88	8651.5	4	278	111	312	301,300
PENNSYLVANIA																									
Pa. P. & L. Co. System	41	959.5	36	848	4	110	1	1.5	22	1254	14	1026	3	213	5	15	145	5328.5	113	3596.5	24	1707	8	25	1,680,000
Johnstown	2	19.5	2	19.5					4	325	4	325					6	344.5	6	344.5					250,000
Philadelphia	176	4198.72	132	3301.15	11	841.1	33	56.47	257	8656.95	183	7310.65	16	1279	58	67.3	843	34820.67	571	27812.8	110	6740.1	162	267.77	2,754,000
Pittsburgh	118	2370	88	2295	4	33.5	26	41.5	212	4095.25	165	3988	4	49.5	43	57.75	515	12069.75	391	11241.25	17	681	107	147	1,400,000

[Continued on next page. See footnotes on next page]

[Continued from preceding page]

Area	1936 All Bldgs. No.	1936 All Bldgs. Hp	1936 Com. & Inst. No.	1936 Com. & Inst. Hp	1936 Industrial No.	1936 Industrial Hp	1936 Residential No.	1936 Residential Hp	1937 (11 Months) All Bldgs. No.	1937 (11 Months) All Bldgs. Hp	1937 (11 Months) Com. & Inst. No.	1937 (11 Months) Com. & Inst. Hp	1937 (11 Months) Industrial No.	1937 (11 Months) Industrial Hp	1937 (11 Months) Residential No.	1937 (11 Months) Residential Hp	Totals as of Dec. 1, 1937 All Bldgs. No.	Totals as of Dec. 1, 1937 All Bldgs. Hp	Totals as of Dec. 1, 1937 Com. & Inst. No.	Totals as of Dec. 1, 1937 Com. & Inst. Hp	Totals as of Dec. 1, 1937 Industrial No.	Totals as of Dec. 1, 1937 Industrial Hp	Totals as of Dec. 1, 1937 Residential No.	Totals as of Dec. 1, 1937 Residential Hp	Approx. Population of Area
RHODE ISLAND																									
Providence	20	348	17	344			3	4	20	327	19	319			1	8	61	1515	55	1498			6	17	300,000
SOUTH CAROLINA																									
Columbia	8	107.75	8	107.75					14	369.25	14	369.25					26	1101	26	1101					72,000
TENNESSEE																									
Tenn. E. P. Co. System	57	965.28	44	949.2			13	6.08	41	1017.75	33	1011.15			8	6.6	184	3663.98	137	3470	10	84.75	37	49.23	500,000
TEXAS																									
Beaumont & Pt. Arthur	12	786.5	10	745			2	41.5	17	363	15	356	1	4	1	3	46	1726.5	38	1634.5	2	12	6	80	115,000
El Paso	16	153	14	149				4	466	429.75	16	307.75			450	122	525	1349.75							100,000
Houston	107	1614.36	78	1530.36			29	84	223	2860.47	179	2722.89			44	137.58	451	11590.13	357	11291.9			94	298.23	400,000
San Antonio	38	856.1	26	831.6			12	24.5	53	2309.25	41	2281			12	28.25	170	8691.35	121	8534.35	2	67.25	47	89.75	231,500
VIRGINIA																									
Va. E. & P. Co. System	63	1339	*	*	*	*	*	*	64	796	*	*	*	*	*	*	247	7309	*	*	*	*	*	*	394,000
Lynchburg	21	40	21	40					8	33.5	7	33			1	0.5	33	163	30	150	1	7.5	2	5.5	50,000
Va. P. S. Co. System	29	99	*	*	*	*	*	*	50	572	*	*	*	*	*	*	*	*	*	*	*	*	*	*	300,000
WASHINGTON																									
Spokane & vicinity	7	65	6	64			1	1	12	287	12	237					29	2441	28	440	100	2000	1	1	250,000
WISCONSIN																									
Milwaukee	29	622.45	24	610.25	*	*	5	12.2	48	1180.7	40	1168.45	*	*	8	12.25	138	5869.7	114	5313.35	*	*	24	56.35	578,000
Racine	2	3.5					2	3.5	2	85.5	1	5.5	1	80			4	89	1	5.5	1	80	2	3.5	70,000

responsive to air conditioning sales this year. The replies, when combined, resulted in the following:

First choice ..... Commercial Stores
Second choice ..... Restaurants
Third choice ..... Private offices
Fourth choice ..... Theaters
Fifth choice ..... Hotels

Readers are cautioned not to read into this analysis anything more than it purports to be—a cross section of general opinion relative to general market opportunities.

According to William B. Henderson, executive vice-president of the Air Conditioning Manufacturers' Association, air conditioning foresees in 1938 its biggest year. The industry attained new highs in 1937, installed cost of equipment sold by members of the A. C. M. A. exceeding by 70 per cent their sales in 1936.

Footnotes for Table of Air Conditioning Installations

*Asterisk indicates figures not available. Figures include connected horsepower for fans, pumps, etc., as well as compressors unless otherwise stated in following footnotes. Figures do not include installations not on public utility company's lines unless otherwise stated in following footnotes. Figures preceded by (a) in following notes are approximate capacities of installations as of Dec. 1, 1937, *not* having motor driven refrigeration machines (steam jet, well water, etc.), when information is available.

Connecticut, Hartford—(a) 77 tons.

District of Columbia, Washington—32 tons well water and adsorption. Data on commercial and institutional installations include following governmental buildings: 1936—6 jobs, 5563 hp; 1937—17 jobs, 2262.5 hp; total as of Dec. 1, 1937—81 jobs, 27,376.75 hp.

Illinois, Chicago—Figures include jobs not on utility's lines.

Indiana, Indianapolis and Marion County—Figures include jobs not on utility's lines. (a) 600 tons.

Iowa, Des Moines—There are no installations not on utility's lines in this territory, utility reports. (a) "Small percentage of total."

Kentucky, Louisville—Figures include jobs not on utility's lines.

Maryland, Baltimore—(a) 360 tons steam and 200 tons adsorption on reporting utility's steam and gas lines.

Massachusetts, Boston Edison Co.—(a) Less than 1 per cent.

Michigan, Detroit—Connected horsepower for fans and pumps not included; figures given are for compressor hp only.

Michigan, lower except Detroit—Figures include jobs not on utility's lines. (a) 3497 tons.

Minnesota, Minneapolis—(a) 1937 figures include 34 jobs with 813 hp using deep well or city water; figures as of Dec. 1, 1937, include 180 such jobs with 2452 hp.

Missouri, Kansas City—Figures include jobs not on utility's lines. (a) 1200 tons. In addition to figures tabulated, there is a total of 547.3 hp for evaporative cooling installations.

Nebraska, Omaha—(a) 40 tons.

New Hampshire, Manchester—Figures include jobs not on utility's lines. (a) 6 tons.

New York, New York City—The total horsepower of the air conditioning installations in Manhattan and The Bronx prior to 1935 is 41,838 hp, which includes electric, gas, and steam, but no private plants. The total hp of the electric, gas, and steam air conditioning installations placed on utility's lines prior to 1937 is 12,435 in Brooklyn, and 5111 in Queens. The number of air conditioning installations in Manhattan, The Bronx, Queens, and Brooklyn prior to 1937 including equipment operating on private plants is 1292, with a total connected load of 84,379 hp. The total connected number of electric air conditioning installations placed on utility's lines in Manhattan, The Bronx, Queens, and Brooklyn during 1937 is 415, with a connected load amounting to 16,203.5 hp, not including private plants.

New York, Syracuse—(a) 100 hp.

Ohio, Cincinnati—(a) 500 tons.

Ohio, Dayton—(a) 300 tons.

Ohio, Toledo—(a) 400 tons, 40 of which are 1937 jobs. Figures include jobs not on utility's lines.

Oklahoma, Oklahoma Gas and Electric Co.—In addition to figures tabulated there is a total of 1670 hp for 193 evaporative cooling systems.

Oklahoma, Public Service Co. of Oklahoma Territory, Tulsa—(a) 350 tons.

Pennsylvania, Philadelphia—Figures include jobs not on utility's lines. (a) 740 tons.

Texas, El Paso—(a) 1937 figures include four installations totalling 178.5 hp *with* refrigeration; balance are evaporative, air washer, etc.

Texas, San Antonio—(a) 50 hp.

Wisconsin, Milwaukee—Figures cover only comfort air conditioning installations using mechanical refrigeration.

Wisconsin, Racine—(a) 10 tons cold water.

SHOW SECTION

At the Grand Central Palace

JANUARY 24-28

New York

Industry's Big Week in New York.....Page 30

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AT THE BILTMORE
Forty-fourth Annual Meeting of the
American Society of
Heating and Ventilating Engineers

AT THE ROOSEVELT
Annual Meetings of the
American Society of
Refrigerating Engineers
and the
National Warm Air Heating and Air
Conditioning Association

Annual Society and Association Meetings

Exposition

Make

Industry's Big Week in New York—January 24-28

THE air conditioning, heating and refrigeration industries get together in important annual technical and business sessions and also go on display in handsome and diversified fashion in New York the last week of January. The "oneness" of their interests is shown in the joint sessions scheduled between the organizations representing each industry and in the completeness and wide range of exhibits of manufacturers serving them all.

At the Grand Central Palace some 300 manufacturers are displaying their products in the 5th International Heating, Ventilating and Air Conditioning Exposition.

At the Biltmore the American Society of Heating and Ventilating Engineers, for the 44th year, is presenting technical papers on subjects of current, vital importance to engineering design and practice in this field. A glance at the program on a following page reveals the selection which will be presented and discussed on the meeting floor.

Similarly, the American Society of Refrigerating Engineers and the National Warm Air Heating and Air Conditioning Association will be separately debating and discussing their particular interests and, on different days, getting together with the American Society of Heating and Ventilating Engineers to study matters of common concern. Programs of the meetings of these two organizations are also presented on a following page.

The exposition is sponsored by the American Society of Heating and Ventilating Engineers and its purpose, in the words of the management, is "to serve as a preview to progress in the heating, ventilating and air conditioning industries. It is a clearing house for ideas and developments in these fields. Exhibits portraying the latest technical advances in the several related fields offer a basis for comparison and exchange of methods. In its previous appearances this exposition has provided the groundwork for future trends."

The displays will fall into the general product classifications of combustion apparatus, hydraulic apparatus, steam and hot water heating equipment, ventilating and air conditioning apparatus, control appliances, refrigeration equipment, insulation, metals and alloys. Admission will be by invitation or by badges of Society members. With nearly 28,000 visitors registered at the last exposition in Chicago it is obvious that it is an event of outstanding significance and importance.

A complete list of exhibitors, the products they will display, their representatives in attendance and floor plans of the exhibit space are all contained in this section.

The Meetings

The programs give detailed information on each meeting. It is evident from them that educational and informative opportunities abound. In the sessions of the American Society of Heating and Ventilating Engineers, technical discussions will center around the broadening applications of air conditioning to the treatment of disease, physiological reactions of people under varying atmospheric conditions, requirements for correct summer cooling and standards in the performance of air conditioning, heating and ventilating equipment.

On the lighter side, recreation and opportunities for contacts and sociability have been just as carefully arranged. The New York Chapter of the American Society of Heating and Ventilating Engineers is playing host in lavish fashion.

On Monday night there is a "get-acquainted" dinner. On Tuesday night there is an International Casino party. On Thursday night is the annual banquet. An inspection trip of the new Lincoln Vehicular Tunnel is arranged for Thursday afternoon. An alternate inspection trip to the Beechnut plant is offered the same afternoon.

As for the ladies, shopping tours, theater parties, bridge and sight-seeing trips will keep them pleasantly occupied between times.

It all adds up to a big week and tribute should be paid to the hard-working New York arrangements committee:—A. C. Buensod, general chairman; Alfred Engle, vice chairman; A. J. Offner, chairman, inspection and transportation; G. E. Olsen, chairman, entertainment; H. W. Fiedler, chairman, ladies committee; Russell Donnelly, chairman, banquet; F. E. W. Beebe, chairman, finance; W. W. Timmis, chairman, reception; C. H. B. Hotchkiss, chairman, publicity.

THE WEEK AT A GLANCE

EVENTS: 5th International Heating, Ventilating and Air Conditioning Exposition.

44th Annual Meeting, American Society of Heating and Ventilating Engineers.

33rd Annual Meeting, American Society of Refrigerating Engineers.

25th Annual Convention, National Warm Air Heating and Air Conditioning Association.

PLACE: New York City.

Exposition—Grand Central Palace

A.S.H.V.E.—Hotel Biltmore

A.S.R.E.—Roosevelt Hotel

N.W.A.H.A.C. Assn.—Roosevelt Hotel

TIME: Exposition—Opens 2:00 p. m., Monday, January 24, and 12 Noon other days. Closes 10:30 p. m. each evening. January 24-28.

A.S.H.V.E.—Registration, committee meetings Monday, January 24, 9:30 a. m. Technical Sessions: Tuesday, Wednesday, Thursday and Friday, January 25, 26, 27 and 28. Joint session A.S.H.V.E.-N.W.A.H.A.C. Assn. Tuesday, 2:00 p. m. Joint session A.S.H.V.E.-A.S.R.E., Wednesday, 2:00 p. m.

A.S.R.E.—Technical sessions, Tuesday, Wednesday and Thursday, January 25, 26 and 27. Joint meeting with A.S.H.V.E. Wednesday, 2:00 p. m.

N.W.A.H.A.C. Assn.—Committee and board meetings Monday, January 24. Technical and business sessions Tuesday and Wednesday, January 25 and 26. Joint session with A.S.H.V.E. Tuesday, 2:00 p. m.

Programs for the Meetings

American Society of Heating and Ventilating Engineers—Hotel Biltmore

44th Annual Meeting

Sunday, January 23

12:30 p. m.—Council Luncheon and Meeting.

Monday, January 24

9:30 a. m.—Registration (Fountain Court, 19th floor, Hotel Biltmore).

10:00 a. m.—Conference of Chapter Delegates (Cafe Moderne, 19th floor).

2:00 p. m.—Nominating Committee (Cafe Moderne, 19th floor).

2:00 p. m.—Opening of Heating and Ventilating Exposition, Grand Central Palace.

6:30 p. m.—Get-Acquainted Dinner (Bowman Room). President's Reception.

Tuesday, January 25

9:00 a. m.—Registration—(Fountain Court, 19th floor, Hotel Biltmore).

9:30 a. m.—Greetings—W. E. Heibel, President, New York Chapter.
Response—President D. S. Boyden.
Reports of Committees.
Technical Papers—
Application and Economy of Steam Jet Refrigeration to Air Conditioning, by A. R. Mumford and A. A. Markson.
Cooling Tower Equipment and Its Relation to Water Conservation, by S. I. Rottmayer.
Report of Tellers of Election.

10:30 a. m.—Ladies' Shopping Tour.

12:00 noon—Heating and Ventilating Exposition—Grand Central Palace.

2:00 p. m.—Joint session with National Warm Air Heating and Air Conditioning Association (Hotel Biltmore Ballroom, 19th floor).
Technical Papers—
Fundamentals Developed from Twenty Years of Research by the National Warm Air Heating and Air Conditioning Association, by A. P. Kratz.
Air Distribution from Side Wall Outlets by D. W. Nelson and D. J. Stewart.
Condensation Within Walls by F. B. Rowley, A. B. Algren and C. E. Lund.

2:30 p. m.—Ladies' Bridge (Empire Room, Hotel Biltmore).

10:00 p. m.—International Casino Party (45th St. and Broadway).

Wednesday, January 26

9:30 a. m.—Technical Papers (Ballroom, 19th floor, Hotel Biltmore).
Studies on Bacterial Control in Air Conditioning, by T. S. Carswell, J. D. Fleming and H. K. Nason.
Physiologic Response of Man to Environmental Temperature, by Dr. F. K. Hick, Dr. R. W. Keeton and Nathaniel Glickman.
Control of Body Heat Loss through Radiant Means, by Dr. C. A. Mills and Dr. Cordelia Ogle.

10:30 a. m.—Ladies' Tour of Radio City and NBC Studios.

12:00 noon—Heating and Ventilating Exposition, Grand Central Palace.

2:00 p. m.—Ladies' Theater Party.

2:00 p. m.—Joint Session with American Society of Refrigerating Engineers (Hotel Biltmore Ballroom, 19th floor).
Technical Papers—
Control of Air Conditioning in Large, Medium and Small Buildings, by W. E. Zieber and S. F. Nichol.
Physiological Reactions and Sensations of Comfort under Various Atmospheric Conditions, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge.
Performance Tests of Asbestos Insulating Air Duct, by R. H. Heilman and R. A. MacArthur.

6:30 p. m.—Past Presidents' Dinner (Room 114, Hotel Biltmore).

Thursday, January 27

9:30 a. m.—Technical Papers (Ballroom, 19th floor, Hotel Biltmore).
Effects of Artificial Lighting on Air Conditioning by Walter Sturrock.
Comparative Analysis of Office Building Air Conditioning Systems, by J. R. Hertzler.
Ventilating the Lincoln Vehicular Tunnel, by C. W. Murdock.

10:30 a. m.—Trip of Inspection *S.S. Rex* for Ladies.

12:00 noon—Heating and Ventilating Exposition, Grand Central Palace.

12:30 p. m.—Luncheon Meeting, Advisory Exposition Committee, Chemists Club.

12:45 p. m.—Ladies' Theater Party, Radio City Music Hall.

2:00 p. m.—Inspection Trip, Lincoln Vehicular Tunnel; Inspection Trip, Beechnut Plant.

7:00 p. m.—Annual Banquet (Ballroom, Hotel Biltmore).

Friday, January 28

9:30 a. m.—Technical Papers (Ballroom, 19th floor, Hotel Biltmore).
Draft Temperatures and Velocities in Relation to Skin Temperature and Feeling of Warmth, by F. C. Houghten, Carl Gutberlet and Edward Witkowski.
Study of Methods of Control and Types of Registers as Affecting Temperature Variations in the Research Residence, by A. P. Kratz and S. Konzo.
Summer Cooling Requirements of 275 Workers in an Air Conditioned Office, by A. B. Newton, F. C. Houghten, Carl Gutberlet and R. W. Qualley.

12:00 noon—Council Meeting (Cafe Moderne, 19th floor, Hotel Biltmore).

12:00 noon—Heating and Ventilating Exposition, Grand Central Palace.

American Society of Refrigerating Engineers—Roosevelt Hotel

33rd Annual Meeting

Tuesday, January 25

9:00 a. m.—Registration opens.

10:00 a. m.—First Session. Topic: Refrigeration of Foods. Guest Chairman: George A. Horne.

Introduction of Chairman, by President H. M. Williams.

Refrigeration in Modern Breweries, by Geo. B. Bright.

The Field of the Food Technologist, by B. E. Proctor.

How Cities Are Fed, informal talk by Walter P. Hedden, Chief, Bureau of Commerce, Port of New York Authority.

12:30 p. m.—Welcome Lunch. Toastmaster: Gardner Poole. Speaker: Johnson O'Connor, "Engineering Humanics."

2:30 p. m.—Second Session. Topic: Fluid Flow. Chairman: Crosby Field.

Flow of Refrigerant Fluids, by A. B. Stickney.

Viscosity of Oils Diluted with Refrigerants, by O. C. Rutledge.

Industrial Relations, informal talk by L. A. Ruthenberg.

1:00 p. m.—Luncheon and Bridge for Women Guests at the Roosevelt.

9:30 p. m.—Night Club Party, International Casino, jointly with A.S.H.V.E. and N.W.A.H.A.C.A.

Wednesday, January 26

10:30 a. m.—Third Session. Topic: Theoretical Questions. Chairman: Harry D. Edwards.

Heat Transfer in Low Temperature Brine Coolers as a Function of Velocity and Viscosity, by J. C. Consley.

New Insulation Studies, by F. G. Hechler and E. R. Queer.

Statistics in Refrigeration, by David L. Fiske.

The Thermodynamics of the Absorption Refrigerating Machine, by B. H. Jennings and F. P. Shannon.

12:30 p. m.—Luncheon for Section Delegates.

2:30 p. m.—Fourth Session, jointly with the A.S.H.V.E. at Hotel Biltmore. Chairman: Willis H. Carrier.

Control of Air Conditioning in Large, Medium and Small Buildings, by W. E. Zieber and S. F. Nichol.

Performance Tests of Asbestos Insulating Air Duct, by R. H. Heilman and R. A. MacArthur.

Physiological Reactions and Sensations of Comfort under Various Atmospheric Conditions, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge.

7:30 p. m.—Annual Dinner Dance.

Thursday, January 27

10:30 a. m.—Fifth Session. Topic: Refrigerating Machinery. Chairman: H. M. Williams.

Refrigeration Storage Applied to Air Conditioning, by W. F. Friend.

Report of Joint Committee on Air Conditioning Standards.

Effect of Superheat on Condensing Unit Efficiency, by L. W. Child.

Extended Surface Cooling Units—Design Calculations, by R. H. Swart.

Trade Schools in Refrigeration, informal talk by Dr. J. S. Noffsinger, National Home Study Council.

Introduction of New Officers.

12:30 p. m.—Council Luncheon.

2:30 p. m.—Inspection of Lincoln Tunnel.

National Warm Air Heating and Air Conditioning Association—Roosevelt Hotel

25th Annual Meeting

Monday, January 24

Committee and Board of Directors meetings.

Tuesday, January 25

9:30 a. m.—Message by President W. L. Rybolt (Grand Ballroom, Roosevelt Hotel).

Address by Dr. A. C. Willard.

The Legislative Strait Jacket on Business, by Emanuel D. Celler.

Value of Cooperation Between the Architect and Our Industry, by C. A. Kissinger.

What's Ahead in Residence Building, by Thomas S. Holden.

2:00 p. m.—Joint session with A.S.H.V.E. (Hotel Biltmore Ballroom, 19th floor).

Fundamentals Developed from Twenty Years of Research by the National Warm Air Heating and Air Conditioning Association, by A. P. Kratz.

Air Distribution from Side Wall Outlets, by D. W. Nelson and D. J. Stewart.

Condensation Within Walls, by F. B. Rowley, A. B. Algren, and C. E. Lund.

9:30 p. m.—Night Club Party, International Casino, jointly with A.S.H.V.E. and A.S.R.E.

Wednesday, January 26

9:30 a. m.—Research Session. Chairman: C. A. Olsen.

Report of Research Advisory Committee, by F. G. Sedgwick.

Report on summer cooling, furnace sizes, heat losses and temperature drops in forced air heating ducts, by S. Konzo and R. B. Engdahl.

1:30 p. m.—Installation Codes Committee Report, by J. D. Hoffman.

Recent Developments in the Filtration of Air, by H. E. Birkholz.

Meeting Outside Competition or Comparative Values, by Frank E. Mehrings.

Annual Election of Officers.

[Note—These programs were compiled from the latest information available from the headquarters offices of the respective societies at the time of going to press.]

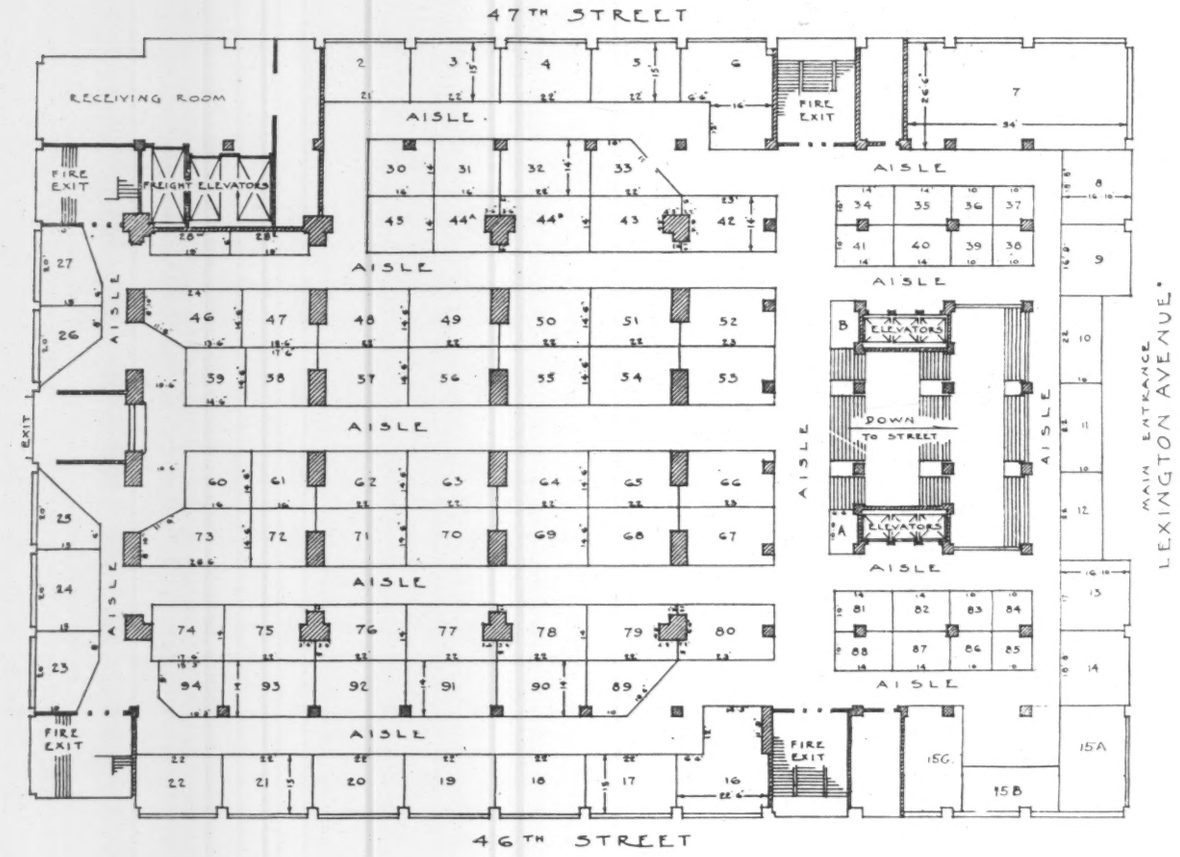
List of Exposition Exhibitors and Their Exhibits

On these pages will be found the names of manufacturers who are exhibiting at the Exposition, their booth numbers, the names of representatives who are in attendance and a condensed list of the products being exhibited. Those who desire to locate any manufacturer's booth will find the exhibitor's name listed in its proper alphabetical place and opposite it his booth number. The diagram of the exhibit hall is on another page showing, by number, the location of each booth.

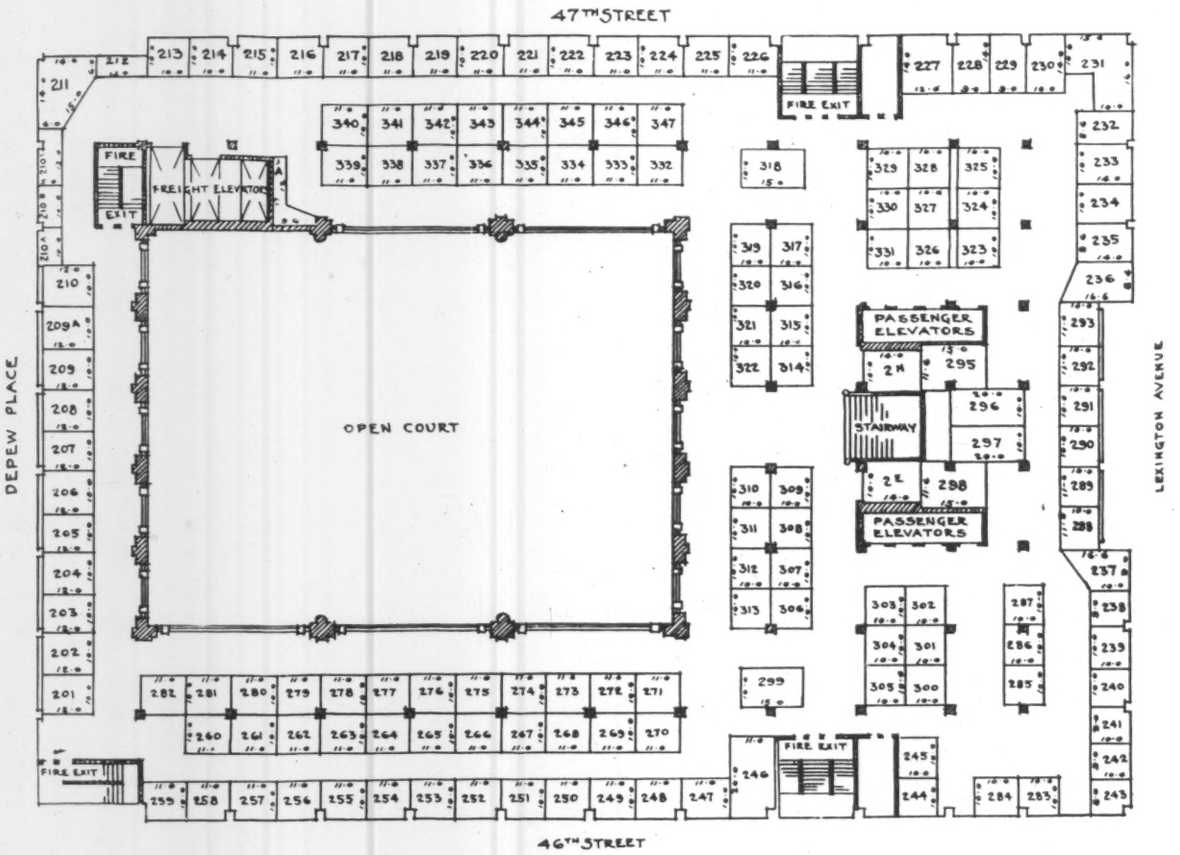
Any additional information will be gladly supplied by HEATING, PIPING AND AIR CONDITIONING at Booth 42. Available advance information on products being shown for the first time at this show as well as information on operating exhibits, are evidence that those who are particularly interested in the latest developments and in construction and operating details will find plenty of opportunity to see them. Improved, modern design is the feature of every exhibit and inspection of every booth is recommended.

NAME OF EXHIBITOR	BOOTH NO.	REPRESENTATIVES IN ATTENDANCE	PRODUCTS ON EXHIBITION
Adams Engineering Co., Chicago, Ill.	550–551		
Advance Engineering Co., New York, N. Y.	288–289		See Armstrong Machine Wks.
Aerofin Corp., Syracuse, N. Y.	230		Heating and cooling surface
Air Conditioning Blue Book, Chicago, Ill.	11		Books and Publications
Air Conditioning combined with Oil Heat	83–84		Publication
Air Conditioning Distributors Co., New York, N. Y.	454	C. W. Walter	See Apex Electrical Mfg. Co.
Air Conditioning Trends, Chicago, Ill.	42	See Keeney Publishing Co.	Publication
Air Controls, Inc., Div. of Cleveland Heater Co., Cleveland, O.	316–317	B. G. Krause, G. L. Schwarz, G. A. Krauss, R. B. Thompson, C. E. Rainsbury	Blower-filter units; blowers; ventilating equipment; furnace casings
Air Devices Corp., (Thermal Units Mfg. Co.), Meriden, Conn.	535	R. J. Schumacher	Unit heaters; automatic humidifiers; safety heat fan
Air-Maze Corp., Cleveland, O.	528	F. M. Paul, O. W. Bochat, E. B. Treidler	Permanent type, cleanable air filter panels; panel holding frames; all-purpose type air filters
Airtemp, Inc., A Subsidiary of Chrysler Corp., Dayton, O.	70	I. C. Baker, C. R. Neeson, R. G. Wyld, R. A. Gonzales, Roy M. Miller	Radial compressors; 3-ton "All-In-One" conditioners; self-contained room units; direct-fired winter conditioners for oil or gas; conversion oil burners
Airtherm Manufacturing Co., St. Louis, Mo.	344–345	F. P. Kohlbry, R. J. Tenkonohy, C. F. Broeder	Unit heaters
Alco Valve Co., Inc., St. Louis, Mo.	248–249	A. B. Schellenberg, J. L. Shrode, A. Ulbert	Thermo valves, expansion valves, magnetic valves, float valves, float switches for refrigerant controls
Aldrich Co., Peoria, Ill.	463–464	L. I. Aldrich, N. E. Aldrich, Carl O. Staley, Alan H. Townsend, A. A. Bauer, C. W. Cope	7 models of domestic oil burners
Allis-Chalmers Mfg. Co., Milwaukee, Wis.	203–204–205		Motors, pumps, V-belt drives
Alpha Steam Specialty Co., New York, N. Y.	528	O. W. Bochat, F. M. Holbrook, Philip Miller, C. Ahrens	N. Y. representatives for Air-Maze Corp., Richmond Engineering Co., and Schade Valve Co.
American Air Filter Co., Inc., Louisville, Ky.	451–452–453	H. C. Murphy, J. R. McConnell, H. E. Birkholz, J. F. Reed, W. C. Kruse, J. H. Hucker, P. L. Pryibil, P. F. Chartres, F. C. Clifford, J. M. Pettingell, R. A. Clapp, Jr., F. F. Espenschied, G. A. Stevens, A. K. Howell, C. H. Hills	Armored multi-panel filters; automatic filters; permanent and throw-away filters; new electrostatic air cleaners
American Artisan, Chicago, Ill.	42	See Keeney Publishing Co.	Warm air heating, residential air conditioning and sheet metal publication
American Blower Corp., Detroit, Mich.	3	E. W. Legier, J. I. Frank, E. H. Taze	Air conditioners and unit heaters
American Brass Co., Waterbury, Conn.	65		Copper tubes with wrought copper and cast bronze fittings for heating, refrigeration and air conditioning lines; copper sheet; silicon bronze for ducts and tanks
American Gas Products Corp., New York, N. Y.	5		Gas-fired furnace
American Machine & Metals, Inc., De Bothezat Ventilating Equipment Div.	215		See De Bothezat Ventilating Equipment Div.
American Metal Hose Branch, American Brass Co., Waterbury, Conn.	65	J. P. Meehan, H. J. Chase, W. B. Pickard, C. E. Woodward	Flexible metal hose and tubing; vibration eliminators
American Oil Burners & Heating Utilities, Brooklyn, N. Y.	214	Daniel L. Finke, C. Christofferson, H. Hammer, R. Hammer, K. Vollbrecht	Oil burners, heat extractors, boiler superchargers, boiler generators
American Radiator Co., New York, N. Y.	2 to 6		Boilers, radiators, air conditioners
American Rolling Mill Co., Middletown, O.	69	G. F. Ahlbrandt, J. F. Bryan, T. I. Byrd, W. A. Kuenzel, R. C. Cunningham	Galvanized sheets; galvanized paintgrip sheets; zincgrip sheets; stainless steel sheets; enameling iron sheets; electrical steel sheets; alloy coated sheets.

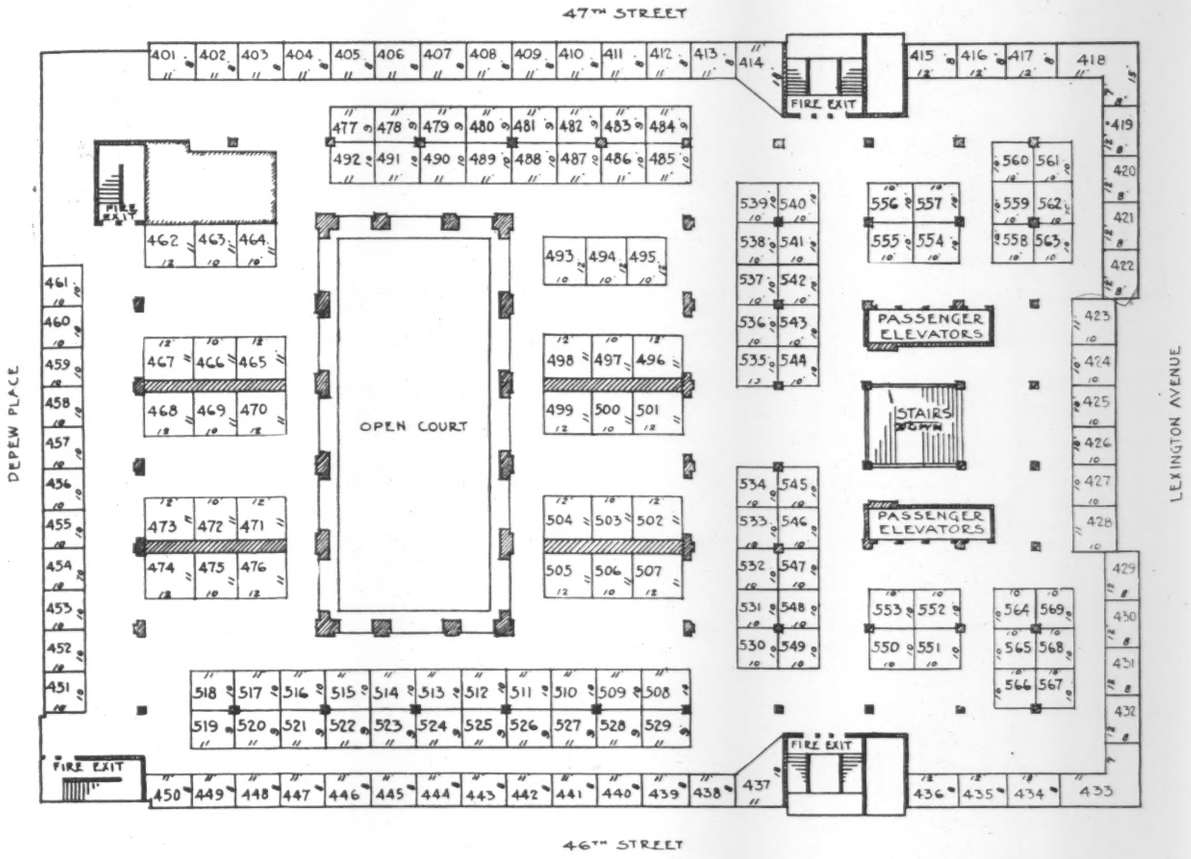
Name of Exhibitor	Booth No.	Representatives in Attendance	Products on Exhibition
American Society of Heating & Ventilating Engineers, New York, N. Y.	A-Main Floor	Society staff	Society activities
American Steel and Wire Co., Chicago, Ill.	52–53		See U. S. Steel Corp.
Anchor Post Fence Co., Baltimore, Md.	508–509–510*		See Fluid Heat Division
Anderson Products, Inc., Cambridge, Mass.	235	L. Lasher, T. R. Lewis, K. G. Agnew, K. B. Gerrish, C. T. Tomlinson	Air valves
Anemostat Corp. of America, New York, N. Y.	2N and 295		High velocity air diffuser; telephone booth ventilating device
Anthracite Industries, Inc., New York, N. Y.	237 to 245, 283 to 287, 300 to 305		Anthracite coal-burning products
Apex Electrical Mfg. Co., Cleveland, O.	454	C. W. Walter, W. C. Mumford, D. O'Neill, C. Hartmann	Humidifiers and air conditioners
Armstrong Cork Products Co., Lancaster, Pa.	274		Cork insulation
Armstrong Machine Works, Three Rivers, Mich.	288–289	O. E. Ulrich, A. E. Armstrong, T. H. Rea	Steam traps and humidifiers
Auburn Automobile Co., Air Conditioning Div., Chicago, Ill.	476		Air conditioning furnace
Auer Register Co., Cleveland, O.	505	Geo. G. Auer, Geo. R. Metzger	Registers and grilles
Auto-Heat Corp., New York, N. Y.	543-544	Otto B. Moody, W. B. Chesbrough, Geo. C. Witzel, Wm. H. Reeve, S. A. Weithorn	Conversion type and boiler burner units
Automatic Burner Corp., Chicago, Ill.	21		Oil burners and range burners
Automatic Heat & Air Conditioning, Chicago, Ill.	11		Publication
Automatic Products Co., Milwaukee, Wis.	261–262	R. W. Johnson, E. A. Vallee, A. Carroll, D. Moerick, A. Brandt, D. Tmey	New model solenoid refrigerant valves; new model thermostatic expansion valves; new thermostats; plus a complete line of refrigerant, gas, oil and water solenoid valves, thermostatic expansion valves, automatic heat regulator sets, humidity control equipment, thermostats and electric, manual and thermostatic constant level oil control valves
Automatic Temperature Control, Inc., New York, N. Y.	560	J. T. Midyette, Jr., Walter S. Hood	Temperature and pressure controls; motorized valves for control of flow of steam and liquids; motorized dampers for air conditioning; micro-balanced three point controls for large buildings
Autovent Fan and Blower Co., Chicago, Ill.	34	Tom Brown, C. E. Wandas, R. F. Ruggles, E. V. Loughran	Propeller fans; volume blowers; unit heaters; attic fans; comfort coolers; V-belt drive unit blowers; cabinet ventilators
Babcock & Wilcox Co., Refractories Div., New York, N. Y.	533–534	M. J. Costello, D. Laing, R. J. Powell, G. M. Frederick, C. A. Underwood, J. E. Brinckerhoff.	Insulating firebrick; precast combustion chamber for household furnaces
Baker Ice Machine Co., Inc., Omaha, Neb.	45	C. A. Baker, R. M. Armstrong, E. L. Cook, Theo. Lindeman	Air conditioning and refrigerating equipment; condensers; water coolers
Baldor Electric Co., St. Louis, Mo.	224	E. Ballman, E. Doerr, O. A. Baumann, E. W. Bruce, D. Halliday, R. H. Jackson, K. S. Jacobsen, S. E. Shea, A. Mandel	Electric motors
Baldwin-Hill Co., Trenton, N. J.	442	E. R. Stevens	Building insulation materials
Balloffet Dies and Nozzle Co., Inc., Guttenberg, N. J.	525	L. Chambre, John H. Davis, Fred Johnson	Oil nozzles
Barber-Colman Co., Rockford, Ill.	323–4–5	D. J. Stewart, R. P. Dewey, C. J. Braatz, E. S. Huffman	Electric temperature control equipment; grilles and registers
Barnes & Jones Inc., Boston, Mass.	312–313	Walter E. Barnes, William T. Jones	Modulation radiator valves; packless quick-opening radiator valves; thermostatic return traps; float and thermostatic traps; boiler return traps; replacement thermostatic units for radiator traps
Beaton and Cadwell Mfg. Co., New Britain, Conn.	422		Pressure reducing and regulating valves; relief valves; air valves; floor and ceiling plates
Beaver Pipe Tools, Inc., Warren, O.	28–E		Full line of pipe tools and machines
Bell & Gossett Co., Chicago, Ill.	28–W	E. J. Gossett, C. E. Pullum, R. E. Moore	Heating specialties, such as boosters, relief and reducing valves, water heaters and compression tanks
Bethlehem Steel Co., Bethlehem, Pa.	71	W. F. Kennedy, A. S. Hogan, H. E. Mitchell, W. T. Carpenter, D. F. Taylor, C. W. Ganzell, R. W. Saunders	Galvanized sheets; pipe
Binks Mfg. Co., Chicago, Ill.	490–91–92	C. R. Kranz, A. J. Styshal, L. A. Hamilton, C. F. Starkweather, Herbert Johnson	Spray cooling towers; spray nozzles for water cooling, air conditioning, air washing
Black & Decker Mfg. Co., Towson, Md.	462		Electric tools
Boiler Room Equipment, Inc., New York, N. Y.	415	Worrell H. Holby, Vernon A. Holby, J. Duncan Holby, Fay L. Kinne	Barometric dampers; smoke controls; tempering valves
Breuer Electric Mfg. Co., Chicago, Ill.	282	A. S. Corte	Furnace and boiler vacuum cleaners
Brown Instrument Co., Philadelphia, Pa.	19–20	See Minneapolis-Honeywell Regulator Co.	Indicating, recording and wet and dry bulb circular chart thermometers; flow meters



Main Floor Plan of Exhibit Space



Mezzanine Floor Plan of Exhibit Space



Third Floor Plan of Exhibit Space

Name of Exhibitor	Booth No.	Representatives in Attendance	Products on Exhibition
Bryant Heater Co., Cleveland, O.	66–67	H. N. Mallon, Lyle C. Harvey, Gordon Rieley, A. E. Schwarz, W. R. Lacey, J. N. Crawford, W. E. Stark, J. E. Kinner, J. M. Traugett, C. H. Cummings, D. E. Leslie, P. M. McCoy, V. B. Beam	Gas fired boilers, furnaces, dehumidifiers, humidifiers, unit heaters
Buffalo Forge Co., Buffalo, N. Y.	26–27	C. C. Cheyney	Ventilating fans; unit heaters; unit conditioners
Burnham Boiler Corp., Irvington, N. Y.	233–234	A. E. Bastedo, A. P. Weiss, J. Balter, A. Hastings, M. C. Cummings, W. A. Bruce, R. H. Doscher, W. V. Skidmore, G. I. Cornwall, E. Van Sise, R. Danks, Geo. E. Sealy, Larry Eckfeldt, A. W. Klein	Boilers, radiators, specialties
Carbondale Division, Worthington Pump & Machinery Corp., Harrison, N. J.	413–414		See Worthington Pump & Machinery Corp., Carbondale Div.
Carey Co., Philip, Cincinnati, O.	411–412	R. A. MacArthur, D. L. Bigelow, L. W. Clarke	Insulation
Carnegie-Illinois Steel Corp., Pittsburgh, Pa.	52–53		See U. S. Steel Corp.
Carrier Corp., Syracuse, N. Y.	227–228–229		Air conditioning equipment
Carter Coal Co., Inc., New York, N. Y.	202		
Cashin Co., W. D., So. Boston, Mass.	332–333		Steam specialties
Celotex Corp., Chicago, Ill.	514		Insulation
Century Electric Co., St. Louis, Mo.	314–315	James Larkin, G. Y. Watt, A. L. Smith, G. E. Moe, Geo. H. Pechin, W. J. Nace, W. L. Dively, Oliver S. Imes	Motors for compressors, fans, blowers, oil burners, stokers, pumps, valves, dampers, etc.
Century Engineering Corp., Cedar Rapids, Ia.	268–269–270	M. D. McWilliams, J. A. Lattner	Oil burners; boiler-burner units; furnace-burner units with winter air conditioning; domestic hot water heaters
Chace Co., W. M., Detroit, Mich.	222	Stanley R. Hood, George W. Wright	Thermostatic bi-metals
Chicago Pump Co., Chicago, Ill.	78	M. I. Weil, H. E. Martin, A. D. Baynes, S. K. Steiner	Circulating pumps; condensation pumps; boiler feed pumps; return line vacuum pumps for heating systems; air washer pumps

Name of Exhibitor	Booth No.	Representatives in Attendance	Products on Exhibition
Cole-Sullivan Engineering Co., Minneapolis, Minn.	540	A. D. Cole, H. J. Sewell, E. W. Laudert, S. H. Burt, I. E. Loring, Chas. Martin	Draft governors
Columbia Steel Co., San Francisco, Cal.	52–53		See U. S. Steel Corp.
Condensation Engineering Corp., Chicago, Ill.	443–444		Heat extractors and chimney l[illegible]
Consumers Petroleum Corp., New York, N. Y.	536		Heat regulator; oil burner; fuel oi[illegible]
Cooper & Cooper, Pittsfield, Mass.	237–245		See Anthracite Industries, Inc.
Cork Import Corp., New York, N. Y.	337	Wm. E. Stewart, E. D. Schive, F. G. Cart, Jr., F. L. Mitchell, G. H. Drake	Corkboard insulation; cork cove[illegible] insulation
Crane Co., Chicago, Ill.	54–55	George M. Hoffman, L. G. Lindsay, Clarence J. Stermer, D. R. Nordwall, L. J. Mauget, E. L. Malm, E. W. Darling, Jr.	A complete line of boilers and radiators; oil burning warm air conditioner; oil burner; boiler-burner unit; stoker-boiler unit; winter air conditioning unit; prefabricated duct system
Crowe Name Plate & Mfg. Co., Chicago, Ill.	273	W. W. Barry, I. R. Smith, T. S. Halpin	Decorative metal parts; nameplates; dials; control plates; panels; scales; refrigerator interior trim and parts; small metal cabinets or housings
Curtis Refrigerating Machine Co., St. Louis, Mo.	495	H. C. Morrison, H. J. Nettler, Warren W. Hull, H. J. Campbell, E. S. Lape, W. M. Clow	Regular water cooled air conditioning condensing units; cooling units; evaporative condensers
Cyclone Fence Co., Waukegan, Ill.	52–53		See U. S. Steel Corp.
Davidson Fan Co., Newton, Mass.	515	A. F. Hinrichsen, L. R. Rickards, Chas. Shaw	Fans
Davies Air Filter Corp., New York, N. Y.	539	Chas. Davies, J. Reilley, S. Gordon, A. Feinsilber	Renewable dry air filters; window filter silencer and ventilator
Dayton Rogers Mfg. Co., Minneapolis, Minn.	474	D. A. Rogers, A. Mill, R. V. Taylor, H. S. Skinner	Metal stampings; stamped parts for air conditioning equipment; metal stampings for control devices
Dayton Rubber Mfg. Co., Dayton, O.	334–335	Karl Glanton, T. C. Davis, T. D. Slingman, J. W. Torrant, D. A. MacDougall, H. L. Boger	Cog-belt V-drives, from fractional to 1,000 hp.
DeBothezat Ventilating Equipment Div., American Machine and Metals, Inc., New York, N. Y.	215	A. R. Stephan, A. M. Harp, M. P. Oothout, J. M. Means, J. E. Fulweiler	Fans; blowers; bifurcators; roof ventilators
Delco-Frigidaire Conditioning Corp., Dayton, O.	72–73		Air conditioning equipment
Detroit Lubricator Co., Detroit, Mich.	2	P. S. Russel, A. G. Hann, R. L. Campbell, F. D. Cooper, C. J. Swan, G. E. Cage, R. E. Townsend, A. A. Putt, W. P. Abbott, F. S. Bartlett, W. H. Homeyer, Vincent Ellis, W. H. Krack	Automatic controls for steam, hot water, warm air heating, air conditioning, refrigeration, oil burners, stokers, gas burners
Dole Valve Co., Chicago, Ill.	206	F. M. Thomson, Earl K. Williams, A. W. Studwell, George F. Martin, Richard Brady, L. A. Fisk, M. E. Northrop, W. B. Chapman, Homer C. Perkins, Thomas G. Colter, Richard G. Goldthwait, P. F. Gilboy, M. D. Hunt, E. C. Curtiss, H. J. Ryan, Park S. Hedley, Wm. G. Boales	Air and vacuum valves
Domestic Engineering, Chicago, Ill.	11		Books and publications
Dooley, Walter F., Boston, Mass.	402–403		
Dreis & Krump Mfg. Co., Chicago, Ill.	436		Steel hand, portable, and press brakes
Dresser Mfg. Co., S. R., Bradford, Pa.	477	C. W. Presdee	Pipe fittings, couplings, and pipe-joint repair devices
Dunham Co., C. A., Chicago, Ill.	46–47		Heating systems
Duo-Therm Division, Motor Wheel Corp., Lansing, Mich.	561–562		See Motor Wheel Corp.
Dupont Fuel Oil Burner Co., Inc., Brooklyn, N. Y.	497		Oil burners
Eagle-Picher Sales Co., Cincinnati, O.	485–486		Insulation
Economy Pumping Machinery Co., New York, N. Y.	58–59		Pumps
Electric Air Heater Co., Div., American Foundry Equipment Co., Mishawaka, Ind.	441	T. J. Oliver, M. O. McClellan, R. E. Pequignot	Electric air heaters for domestic and industrial applications
Electric Furnace Man, Inc., New York, N. Y.	237–245		See Anthracite Industries, Inc.
Electrol, Inc., Clifton, N. J.	554–555–556–557		Conversion oil burners; boiler burner units; direct fired units; split system conditioners; room coolers
Electronic Control Corp., Chicago, Ill.	402	F. J. Schaefer, Walter F. Dooley	Stoker and oil burner controls; modulating steam valve controls
Emerson Electric Mfg. Co., St. Louis, Mo.	340–341	V. J. Maurer, L. F. Blough, R. E. Otto, J. A. Rodgers, O. D. Metz, T. J. Egan, E. E. Harwood, R. W. Walter, G. E. Hansman, T. J. Grennan, D. F. Aisenbrey	Motors and fans
Evans Corp., George, Moline, Ill.	565–566	Reid Evans, C. Warfield	Oil burning air conditioning furnace
Fairbanks, Morse & Co., Chicago, Ill.	89–90	J. W. Bostwick, H. L. Bilsborough	Self-contained air conditioners; remote unit; gas engine driven condensing unit; motor driven condensing unit; hopper type domestic bituminous stoker; bin-feed domestic anthracite stoker

Name of Exhibitor	Booth No.	Representatives in Attendance	Products on Exhibition
Fairfield Oil Heating Co., Inc., Greenwich, Conn.	432	J. R. Johnson, J. Richard Johnson	Oil burners, oil gauge, air-conditioning system
Fedders Manufacturing Co., Buffalo, N. Y.	346–347	H. E. Rieckelman, E. R. Walker, Joe Askin, Harry C. Schmidt, Frank A. Haag, E. J. Moran	Air conditioning units; blast heating & cooling surface; unit heaters and coolers; non-frost coils; ice cube making evaporators; condensers (refrigerant); thermostatic expansion valves; automatic expansion valves; constant pressure valves
Field Manufacturing Co., Chicago, Ill.	415	Fay L. Kinne, Vernon A. Holby, Worrell H. Holby	Barometric draft controls and smoke indicators
Fitzgibbons Boiler Co., Inc., New York, N. Y.	64	J. A. Darts, G. E. Olsen, Paul K. Addams, Arthur W. Wilson, Kenneth L. Mytinger	Boiler-burner unit and direct fired conditioner
Fluid Heat Div., Anchor Post Fence Co., Baltimore, Md.	508–509–510	Wm. F. Brannan, A. Blaeser, F. P. Harbin, T. H. Smoot, D. G. Day, F. J. Spath, Geo. J. Dunn, J. W. Stallings	Oil burners; water heaters; winter conditioning furnaces; burner-boiler units
Fox Furnace Co., Elyria, O.	5–6	C. A. Olsen, M. Suppes, G. Boeddener, E. P. Hayes, D. P. Foote, M. J. Brennan, R. Milholland, E. Hendrickson, J. C. Welch, F. Page, C. E. Johnson, J. E. Maynard	Oil, gas, and coal fired air conditioning units; cooling equipment; blower units; air filters; registers and grilles
Frick Co., Waynesboro, Pa.	74	W. H. Aubrey, C. C. Smith, W. O. Kline, J. T. Murphy, H. R. Howell, M. B. Weinberg	Refrigerating and air-conditioning equipment
Friez & Sons, Inc., Julien P., Baltimore, Md.	49	Denis McCormack, R. R. Chappell, K. E. Whitney, Harry W. Fiedler	Instruments for automatic control, indication and recording of air conditions
Fuel Oil Journal, New York, N. Y.	292–293		Publication
Fulton Sylphon Co., New York, N. Y.	252–253	E. D. Rogers, F. C. Reeder, A. W. King, A. W. Shepherd, F. G. Cross, John Dube	Temperature controls
Furblo Co., Hermansville, Mich.	408	I. W. Rowell, C. H. Lighthart, A. R. Meiche, L. W. Gillespie	Blower-filter units; blower wheels and housings
General Controls Co., San Francisco, Calif.	507	W. D. Weeks	Automatic temperature and flow controls
General Electric Co., Air Conditioning Dept., Bloomfield, N. J.	7		Air conditioning equipment
General Electric Co., Schenectady, N. Y.	24		Motors and controls
General Fittings Co., Providence, R. I.	409–410	C. J. Arthur, C. A. Maine, F. J. Paff, H. F. Horton, S. G. Cady, C. R. Bernstrom	Indirect heaters; tankless heaters; built-in boiler units; mixing valves; boiler boosters
General Oil Heating Corp., West New York, N. J.	455		
Genesee Electric Products Co., Rochester, N. Y.	530		
Gilbert & Barker Mfg. Co., Springfield, Mass.	545 to 549		Oil burners
Gilbert & Son, Inc., Harry E., Bridgeport, Conn.	259	Harry E. Gilbert, Carl G. Nelson, C. R. Courtenay, Rudolph Rehm, W. R. Mustor, G. Page Bouis	Refractories
Grinnell Co., Inc., Providence, R. I.	332–333		Pipe fittings; welding fittings; traps; adjustable hangers; automatic sprinklers; unit heaters
Grob Brothers, Grafton, Wis.	470	Benjamin Grob, Theodore Grob	Die making machinery
			Metal floats
Harris & Co., Arthur, Chicago, Ill.	489	George C. Harris	Oil burner specialties
Harsch Co., Inc., H., Maplewood, N. J.	496	Frank Harsch, Frank Hoffman	Air conditioning registers and grilles; damper regulators; furnace pulleys and chain; cabinet top grilles
Hart & Cooley Manufacturing Co., Holland, Mich.	22	J. H. Van Alsburg, D. J. Jones, R. L. Leigh, L. E. Heasley	
Heating & Ventilating, New York, N. Y.	277		Publication
Heating Journals, Inc., New York, N. Y.	83–84		Publication
Heating, Piping and Air Conditioning, Chicago, Ill.	42	See Keeney Publishing Co.	Commercial, large building and industrial heating, piping and air conditioning publication
Heil Co., Milwaukee, Wis.	23	Jos. F. Heil, Julius P. Heil, C. B. Tamm, George Hochstein, George Kuhlman, R. B. Joys, H. F. Pugh	Direct-fired air conditioning unit; split system, including boiler burner unit and conditioner; oil burner
Henry Furnace & Foundry Co., Cleveland, O.	210A–210B–210C	H. S. Sharp, H. F. Curtis, H. Cunningham	Winter air conditioning units for gas-coal-oil; air conditioning pipe and fittings
Henry Valve Co., Chicago, Ill.	431	C. V. Gary	Refrigerant shut-off valves; relief valves; dehydrators; strainers; scale traps; filters; service tools; forged steel pipe fittings
Hershey Machine & Foundry Co., Manheim, Pa.	237–245		See Anthracite Industries, Inc.
Hinrichsen, Inc., A. F., New York, N. Y.	515	A. F. Hinrichsen, D. O'Neill, Chas. Hartmann, Carl Moser, L. R. Rickards, Chas. Shaw	See Davidson Fan Co. and Montgomery Bros.
Hoffman Specialty Co., New York, N. Y.	58–59		Air valves and heating specialties
Hotstream Heater Co., Cleveland, O.	216		Water heaters
Ilg Electric Ventilating Co., Chicago, Ill.	68	P. D. Briggs, J. O. Lennon, W. G. Thomas, P. A. Hoffman, G. Pearce, C. W. Stewart	Blowers; fans; unit heaters, both steam and electric types; cooling units; kitchen ventilating fans; attic fans
Illinois Engineering Co., Chicago, Ill.	81–82	J. C. Matchett, J. L. Ehretsman, R. W. Leutwiler, A. L. Browne, H. Fox	Selective pressure, pneumatic zone, and electric zone control systems; heating specialties
Illinois Testing Laboratories, Inc., Chicago, Ill.	38	M. D. Pugh, J. H. Torrey	Air velocity meters; surface temperature pyrometers; distant indicating electric thermometers
Imperial Brass Mfg. Co., Chicago, Ill.	264		
Independent Air Filter Co., Chicago, Ill.	487–488	H. W. Pound, J. L. Harpham, Hy. Gitterman	Air filters

Name of Exhibitor	Booth No.	Representatives in Attendance	Products on Exhibition
Independent Register Co., Cleveland, O.	319–320	Blair A. Quick, B. N. Richardson.	Air conditioning registers and grilles
Industrial Press, New York, N. Y.	277		Publications
Ingersoll-Rand Co., New York, N. Y.	512–513	E. L. Champion, J. A. Anderson, L. L. Lauver, A. D. Karr, J. H. Metzgar	Water vapor refrigeration; pumps
Insto-Gas Corp., Detroit, Mich.	529	H. F. Belcher, John R. Robinson.	Brazing torches
International Boiler Works., East Stroudsburg, Pa.	237–245		See Anthracite Industries, Inc.
James Regulator Co., Pottsville, Pa.	237–245		See Anthracite Industries, Inc.
Janette Mfg. Co., Chicago, Ill.	475	J. I. Janette, D. W. Smith, R. L. Medbery, S. J. Rogers, A. Nelson	Blower wheels; D.C. to A.C. rotary converters; motorized speed reducers
Jenkins Bros., New York, N. Y.	13		Valves
Johns-Manville, New York, N. Y.	87–88	H. R. Berlin, P. J. Washburn, W. J. Hodge, N. B. Sherrill, J. H. MacKay, H. E. Bowman, W. W. Cullin, T. J. Roberts, F. J. Warem, L. A. Baldwin, C. A. McGinnis, E. Rinehart, H. A. Kieselbach, G. E. Grimshaw, J. F. Stone, J. C. Crawford, R. C. Parlett, G. A. Barker, C. G. Dandrow, C. Hall, E. A. Phoenix, C. G. Bittner, W. P. Gibbons	Insulation for buildings and ducts; pipe covering; packing
Johnson Co., S. T., Philadelphia, Pa.	256–257–258		Oil burners and new winter air conditioner
Johnson Service Co., Chicago, Ill.	75	F. P. Brophy, Joe Wheeler, Jr., P. A. Riccio, H. W. Alyea, A. C. Buffalano	Automatic temperature and humidity control systems for heating, cooling, ventilating, air conditioning
Joliet Heating Corp., Joliet, Ill.	499–500–501		Warm air furnaces
Jones & Laughlin Steel Corp., Pittsburgh, Pa.	48		Sheets; pipe
Kainer & Co., Chicago, Ill.	221		Hot water heating specialties
Kaustine Co., Inc., Perry, N. Y.	469	J. B. Mollnow, Chas. Murdock	Furnaces and dampers
Keeney Publishing Co., Chicago, Ill.	42	F. P. Keeney, W. J. Osborn, R. P. Wettstein, Chas. E. Price, R. A. Jack, C. M. Burnam, Jr., J. D. Wilder, J. D. Thomas	Books and publications
Kelvinator Div. of Nash-Kelvinator Corp., Detroit, Mich.	326 to 331	H. M. McGaughey, E. L. Sylvester, L. G. Estep, B. M. Hanley, L. T. M. Ralston, W. E. MacBeth, Larry Williams	Residential air conditioning system; cut-away view of oil-fired boiler; coal stoker; commercial air conditioning compressor unit and evaporative condenser; suspended type a.c. unit; room coolers
Kewanee Boiler Corp., Kewanee, Ill.	3	Ben H. Schulze, L. W. Charlet, A. P. Hare, J. M. Hartman	Boilers; heaters; coils; tanks
Kleen-Heet, Inc., Chicago, Ill.	21		Oil burners
Korth Oil Burner Corp., Roselle Park, N. J.	526–527	Arnold Eckhart, John L. Lehman, George S. Siggins	Oil burners
Koven & Bros., L. O., Jersey City, N. J.	211		Tanks
Lamneck Products, Inc., Columbus, O.	260	W. S. Somers, E. H. Paul, P. S. Miller, H. R. Allgrove, R. E. Leffler, W. B. Cooper, C. M. Butler	Prefabricated duct and fittings for forced air and air conditioning installation
Lau Blower Co., Dayton, O.	265–266	E. B. Lau, R. H. Sommers, H. A. Lovell	Blower assemblies, wheels, housings; furnace blower package units; unitair blowers; pulleys; furnace casing moulding
Lochinvar Corp., Detroit, Mich.	502–503–504	Charles A. Wagner, J. J. Cheviron, J. L. Gillen, Harry A. Dibble, Lawrence F. Hess, Howard I. Piatt, Howard Baitty, Delbert Kramer, L. W. Kinear, A. Maxwell	Oil burning furnaces and water heaters
Lycoming Mfg. Co., Spencer Heater Div., Williamsport, Pa.	318		See Spencer Heater Div., Lycoming Mfg. Co.
Lynn Products Co., Lynn, Mass.	433	F. H. Van Blarcom, J. F. McGowan, A. M. Parvin, R. L. Dennis, H. V. Jacobsen	Oil burners and heat circulators
Maas & Waldstein Co., Newark, N. J.	434		Enamels and lacquers
Macrae, Inc., Brooklyn, N. Y.	471	James N. Macrae, H. J. Hueller, J. J. Morro, Gilbert L. Brown, Nicholas Wassil, J. Morgan, J. K. Williamson	Oil burner; oil heating boiler-burner unit
Malleable Iron Fittings Co., Branford, Conn.	35	P. R. Besserman, H. Lindberg, Herbert Gillis	Oil burners; oil burning boilers; malleable iron pipe fittings
Marsh Corp., Jas. P., Chicago, Ill.	94	B. Scudder, A. D. Rose, C. Olmsted, R. S. Gaisford, Tom Kenney, Chas. Kelly, W. Chichester, S. C. Kauffman, C. H. Bevington, H. Kucera, J. G. Lawler, T. K. Birrell, L. D. Angell, H. J. Callahan, W. R. Vom Saal, R. H. Monsell	Heating equipment, industrial instruments, gauges, and thermometers
Marsh Tritrol Co., Chicago, Ill.	94	See Jas. P. Marsh Corp.	Temperature controlling equipment
Maurey Mfg. Corp., Chicago, Ill.	406	S. R. Noble, Carl A. Miller	Steel V pulleys and refrigeration fans
May Oil Burner Corp., Baltimore, Md.	8	P. H. Jacobson, Jos. Geartner, Edw. Ryan, Frank Drake, E. E. Yaggy, Jr.	Oil burners and oil burning furnaces and conditioners
McDonnell & Miller, Chicago, Ill.	41	E. N. McDonnell, B. P. Adams, A. W. Shepherd, G. W. Ramsay	Boiler water feeders; low water cutoffs; constant level valves; humidifier float valves

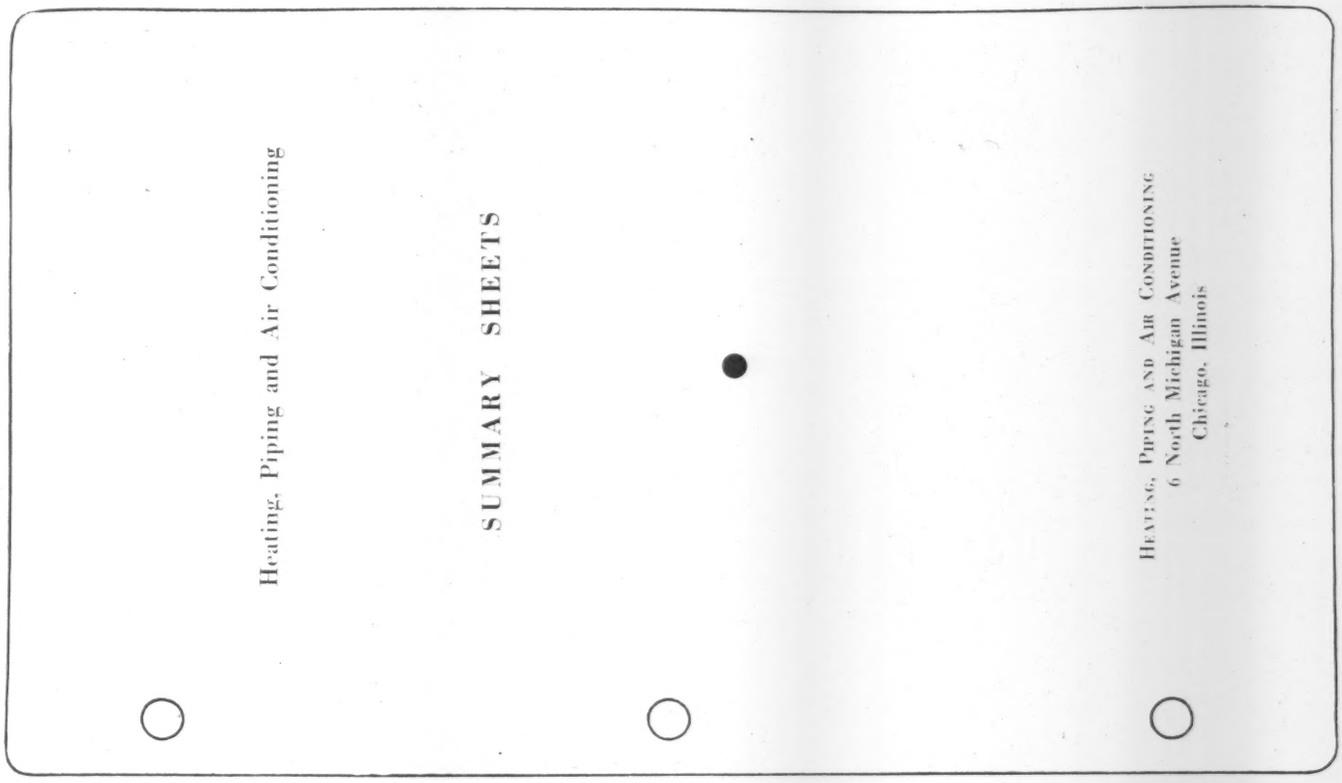
Name of Exhibitor	Booth No.	Representatives in Attendance	Products on Exhibition
Mercoid Corp., Chicago, Ill.	85–86	I. E. McCabe, W. Colterjohn, R. H. Chadwell, J. N. Derby, Jr., W. K. Stauffer, A. W. Barr, K. Reily, A. W. Ayer, M. J. Kirsh, F. W. Harrison, R. B. Powers.	Complete line of automatic controls for oil burners, stokers, refrigeration, air conditioning and industrial application
Meyer & Bro. Co., F., Peoria, Ill.	10	See Meyer Furnace Co.	Furnace pipe and fittings; standardized duct work for air conditioning
Meyer Furnace Co., Peoria, Ill.	10	F. L. Meyer, F. E. Mehrings, E. Eichenberger, O. E. Adams.	Warm air furnaces and domestic air conditioning appliances for coal, oil and gas
Michigan Tank & Furnace Corp., Detroit, Mich.	502–503–504		See Lochinvar Corp.
Micro-Westco Inc., Bettendorf, Ia.	423 to 428	R. F. Valentine, Lee Wernentin, R. Lee, L. S. Knight, L. R. Parrish	Oil burners and pumps
Milwaukee Valve Co., Milwaukee, Wis.	403		Heating specialties
Minneapolis-Honeywell Regulator Co., Minneapolis, Minn.	19–20	H. W. Sweatt, C. B. Sweatt, W. L. Huff, J. W. Pauling, G. D. Kingsland, A. H. Lockrae, Arnold Michelson, E. K. Scoggin, C. D. Lyford, C. W. Nessell, D. J. Peterson, H. E. Chapler, L. B. Miller, W. L. McGrath.	Complete line of temperature, humidity and refrigeration controls
Modine Mfg. Co., Racine, Wis.	93	R. D. Tyler, A. H. Tyler, F. Brissell	Unit heaters and coolers; copper radiation; blast heaters; cooling coils; air conditioner
Mojonnier Bros. Co., Chicago, Ill.	498	H. J. Liedell, D. J. Crumlish, M. D. Bates, H. G. Mojonnier.	Stainless steel water cooling equipment
Monarch Mfg. Wks., Inc., Philadelphia, Pa.	201	J. M. Carroll, E. B. Frame, T. W. Murphy	Oil and water spray nozzles; strainers; pressure regulating valves; air mixing equipment for oil burners
Montgomery Bros., San Francisco, Cal.	515	A. F. Hinrichsen, Chas. Hartmann, Carl Moser.	Ionizing equipment
Motor Equipment Co., Wichita, Kas.	447	J. J. Strang.	Evaporative condensers
Motor Wheel Corp., Duo-Therm Div., Lansing, Mich.	561–562	N. M. Edwards, E. Crary, M. F. Cotes	Winter air conditioning; automatic oil furnace
Mueller Brass Co., Port Huron, Mich.	14		See Streamline Pipe & Fittings Div., Mueller Brass Co.
Mueller Furnace Co., L. J., Milwaukee, Wis.	32–33	H. P. Mueller, E. A. Jones, M. L. LaVorgna, G. A. McDonough, C. L. Hewitt, Jr., R. M. Roseborough, W. H. McKaig, J. G. Werner	Oil, gas, and coal fired air conditioning furnaces; gas-fired forced circulation heater; gas-fired boiler; double radiator coal-fired furnace
Nash Engineering Co., So. Norwalk, Conn.	87–88		Pumps
National Radiator Corp., Johnstown, Pa.	76–77	W. B. Montague.	Heat extractor boilers—hand, stoker, and oil-fired; gas boiler; oil heating unit; commercial steel boilers; convector
National Regulator Co., Division, Minneapolis-Honeywell Regulator Co., Chicago, Ill.	19–20	See Minneapolis-Honeywell Regulator Co.	Pneumatic temperature and humidity controls
National Sheet Metal Contractor, Chicago, Ill.	298		Publication
National Tube Co., Pittsburgh, Pa.	52–53		See U. S. Steel Corp.
Nesbitt, Inc., John J., Holmesburg, Philadelphia, Pa.	15-B	H. Berkley Hedges, T. M. Reed, Jr., D. C. Wiley.	Blast coil heating surface with internal steam distributing tubes
Norge Heating & Conditioning Div., Borg-Warner Corp., Detroit, Mich.	217 to 220		Air conditioning and automatic heating products
Oliver & McClellan, Inc., New York, N. Y.	441		
Owens-Illinois Glass Co., Toledo, O.	481–482		Air filters
Patterson-Kelley Co., Inc., New York, N. Y.	2E	H. P. Wilder, O. S. Pedersen, F. L. Patterson, Jr.	"Freon" water cooler, capacity 25 tons, thermo expansion type
Penn Electric Switch Co., Goshen, Ind.	310–311	Albert Penn, M. E. Henning, R. H. Luscombe, M. Parcaro, D. A. Coon, H. P. Dame, A. L. Rubel, E. J. Corbett, R. V. Clark.	Controls for heating, refrigeration and air conditioning
Perfection Stove Co., Cleveland, O.	15C	C. H. Foulds.	Oil burning, air conditioning heating plants
Perfex Corp., Controls Div., Milwaukee, Wis.	12	Julius K. Luthe, Frederick D. Hansen, Herb. E. Lindemann, Ben L. Boalt, Shepard Barnes, Allen Butler, Chas. B. Soper, C. Ellis Hayes, D. G. Spahr, L. L. Cunningham, W. R. Miller, J. Jones, Homer Malone, J. Deutel	Complete line of automatic controls for oil burners, stokers, and air conditioning
Perfex Corp., Unit Heater Div., Milwaukee, Wis.	39	Gene D. Sickert.	Unit heaters
Petroleum Heat and Power Co., Stamford, Conn.	15-A	Howard W. Dexter and staff.	Oil burners
Petrometer Corp., Long Island City, N. Y.	290–415–416		
Plibrico Jointless Firebrick Co., Chicago, Ill.	40	F. S. Rieder, G. A. Maier, Jerry Galvin	Firebrick and refractory products
Plumbing & Heating News, Chicago, Ill.	41		Publication
Plumbing & Heating Trade Journal, New York, N. Y.	263		Publication
Preferred Utilities Mfg. Corp., New York, N. Y.	254–255	R. S. Bohn, G. P. Gregory, H. E. Lake, W. H. Bohn, J. J. Spalding, Jr., G. Gendall, Jr., M. Sherman, D. Stone.	Oil burner; hot water heater; floatless low water cut-off; oil burner accessories
Propellair Inc., Springfield, O.	441	V. W. Sharpe, T. J. Oliver, M. O. McClellan	Propeller type fans for general ventilation and duct systems
Randall Graphite Products Corp., Chicago, Ill.	342	W. P. Thacher, R. H. Whiteley, M. E. Clark, E. A. Lampe, H. B. Hostetler	Self-aligning, self-lubricating pillow blocks

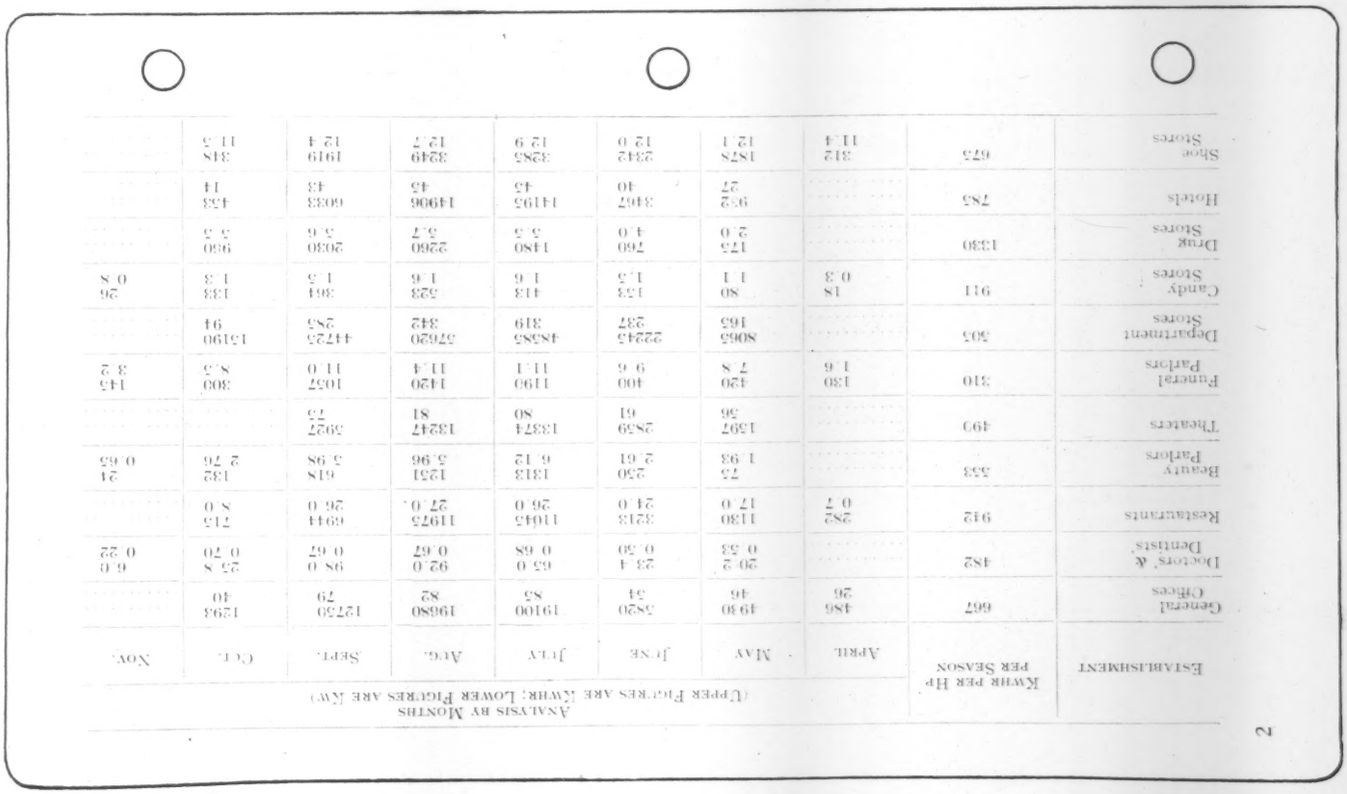
Name of Exhibitor	Booth Nos.	Representatives in Attendance	Products on Exhibition
Refractory & Insulation Corp., New York, N. Y.	559	J. McDonald, J. J. Bliss, F. Christenson, P. Kriegel, H. N. Clark	Oil burner combustion chamb s; hearth, refractory, and insulat g cements; mineral wool home in ulation; plastic insulation for pi es, flanges, boiler walls, etc.
Refrigeration & Air Conditioning Institute, Inc., Chicago, Ill.	445–446		Correspondence and shop trai g course in air conditioning
Refrigeration Appliances, Inc., Chicago, Ill.	494		Coils
Rega Mfg. Co., Rochester, N. Y.	407	C. H. Ager, Wm. McLean	Self-cleaning spray nozzles; w er valve; spray humidifiers
Reif-Rexoil, Inc., Buffalo, N. Y.	417–418	Allan F. Reif, Charles A. Reif, H. J. Schwindt, W. J. Harter, M. J. Gimbrone, Carl Coleman, N. J. Hetherington	Oil burners; boiler burner units; di rect-fired winter air condition g units; stack mounted water h e heaters
Republic Steel Corp., Cleveland, O.	60–61	L. D. Mercer, R. W. Baker, R. H. Sonneborn, N. E. Walker, C. E. Howes, E. A. Regan, R. B. Little	Black and galvanized steel sheets and pipe; furnace casings; wa er heater casings; air conditioner cab nets
Reynolds Corp., New York, N. Y.	516	J. G. Stalb, W. J. Burke	Prefabricated ducts
Richardson & Boynton Co., New York, N. Y.	79		Furnaces
Richmond Engineering Co., Inc., Richmond, Va.	528	E. Hoppe, O. W. Bochat, Philip Miller, Frank Holbrook, E. B. Treidler	Storage type, steam heated hot w er generator
Ric-wiL Co., Cleveland, O.	250–251	H. W. Tucker, Wm. MacLeish	Tile and cast iron conduit for und r-ground steam pipes; waterproof asbestos insulation
Ridge Tool Co., Elyria, O.	483–484	W. W. Graw, C. O. Hollen, W. L. Parcell	Pipe wrenches, pipe cutters, tubing cutters, pipe vises, pipe threaders, threading oil, screw and pipe ex tractors
Ruberoid Co., New York, N. Y.	336		Insulation
Russell & Co., W. A., New York, N. Y.	511	L. E. Pelletier, Fred Gander, W. L. Johnston, W. J. Johnston	Air valves
Sangamo Electric Co., Springfield, Ill.	506	B. W. Dickerman, E. J. Schulenburg	Time switches and timers
Sarco Co., Inc., New York, N. Y.	80		Specialties for low pressure, gravity, vapor, and vacuum steam heating systems, including radiator traps, inlet valves, drip traps, air eliminators, boiler return traps, temperature regulators, etc. System for control of building heating directly by the weather. Thermostatic mixing valves
Schade Valve Mfg. Co., Philadelphia, Pa.	528	F. H. Jones, E. C. Deuter	Steam, water and air reducing valves; float valves; pump governors; rotary type level; control valves; balanced lever valves
Schwitzer-Cummins Co., Indianapolis, Ind.	237–245		See Authracite Industries, Inc.
Scott-Newcomb, Inc., St. Louis, Mo.	531–532	J. P. Feeley	Oil burners; warm air heating and cooling air conditioner; domestic hot water heater; water softeners
Scully Steel Products Co., Chicago, Ill.	52–53		See U. S. Steel Corp.
Seaboard Refractories Co., Perth Amboy, N. J.	259	M. E. Gray	Refractories
Sheet Metal Worker, New York, N. Y.	299		Publication
Silent Glow Oil Burner Corp., Hartford, Conn.	231–232		Oil burners
Simplex Oil Heating Corp., New York, N. Y.	419–420	J. H. Blake, Jr., C. G. Smith, J. G. Kintner, B. E. Sherman, A. L. Bayer	Oil burners
Spencer Heater Div., Lycoming Mfg. Co., Williamsport, Pa.	318		Automatic magazine feed warm air furnace, and boilers. See also Anthracite Industries, Inc.
Spencer Thermostat Co., Attleboro, Mass.	37	V. G. Vaughan, R. M. Glidden, E. P. Jastram, Jr.	Room thermostats; limit controls; stoker timers; thermal relays; miscellaneous disc type thermostatic electric switches; diaphragm type thermostatic gas valves; shutoff valves; pilot burners; gas valves; relief valves; motor protection thermostats
Spoehrer-Lange Co., Clayton, Mo.	448	H. F. Spoehrer, C. C. Grote, C. C. Segrist	Thermostatic expansion valves
Standard Lime & Stone Co., Baltimore, Md.	563	Oliver P. Harris, R. C. Powell, C. L. Reier	Rockwool home insulation materials
Standard Products Co., Steechan Blind Div., Cleveland, O.	465	Fred E. Lawson, Paul L. Pickler	Cordless venetian blind
Staynew Filter Corp., Rochester, N. Y.	449–450	Earl B. Hardy, Eugene J. Cantin	Air filters for ventilation, internal combustion engines and pipe lines
Steel and Tubes, Inc., Cleveland, Ohio	208–209	J. F. Keeler, E. A. Longfellow	Boiler tubes; condenser and heat exchanger tubing; stainless steel tubing
Sterling Engineering Co., Milwaukee, Wis.	212–213		Heating specialties
Stoker & Airconditioner Journal, New York, N. Y.	291		Publication
Streamline Pipe and Fittings Div., Mueller Brass Co., Port Huron, Mich.	14	E. M. Fitzgerald, C. E. Berry	Solder fittings; copper pipe; refrigeration valves; fittings and accessories
Sturtevant Co., B. F., Hyde Park, Boston, Mass.	9	R. H. Keith, W. I. Lawson, R. W. Case, F. G. Gerrish, R. E. Bade, R. A. Benoit, J. F. Munder, Jr., A. C. Bartlett	Fans; motors; unit ventilators; unit heaters.
Sundstrand Engineering Co., Rockford, Ill.	223	W. A. Kemp, W. R. Kiefer, J. H. Hassinger	Oil burners
Sundstrand Machine Tool Co., Rockford, Ill.	343	H. Ekstrom, O. M. Olsen, C. H. Rystrom	Fuel units and oil pumps for pressure atomizing oil burners

Name of Exhibitor	Booth No.	Representatives in Attendance	Products on Exhibition
Supreme Electric Products Corp., Rochester, N. Y.	530	R. J. Barker, Fred W. Armbruster, M. H. Esser, Harry W. Fiedler, J. A. Heller, John G. Foster, Thomas Davenport, H. E. Berendsen, N. A. Glaubach, E. N. Hyde	Electric valves for water, air, oil, gas, steam; spray nozzles; furnace humidifiers; electric magnets; humidity controls; room humidifiers
Surface Combustion Corp., Toledo, O.	278 to 281	R. A. Bell, C. B. Phillips, E. A. Weaver, W. J. Grover, R. L. Towne, F. Hamer, W. W. Boyce, R. Thornburgh, E. Smart, K. G. Hull, D. Preyer	Gas-fired winter air conditioners; gravity heaters; unit heaters; conversion burners; water heaters
Swirling Heat, Inc., Arlington, Va.	404	R. D. Schoenwetter, William J. Debler	Oil burner
Sylphon Control Systems, Inc., New York, N. Y.	252–253		See Fulton Sylphon Co.
Taco Heaters, Inc., New York, N. Y.	44–A		Indirect water heaters; circulators and hot water fittings. See also Anthracite Industries, Inc.
Tennessee Coal, Iron & R. R. Co., Birmingham, Ala.	52–53		See U. S. Steel Corp.
Thermal Units Mfg. Div., Air Devices Corp., Meriden, Conn.	535		See Air Devices Corp.
Thrush & Co., H. A., Peru, Ind.	210	Lewis H. Dietz, C. W. Norby, J. S. Blackmore, R. E. Lund, R. E. Clegg, E. E. Adams, W. D. Carter, Frank C. Bond, Harry E. Thompson	Complete line of high and low pressure reducing valves, relief valves, flow control valves, water circulators, dual control units, and electrical controls
Timken Silent Automatic Div., Timken-Detroit Axle Co., Detroit, Mich.	56–57	Willard F. Rockwell, T. A. Crawford, L. H. Holman, Ralph Hooke, F. M. Jordan, M. B. Ells, T. D. McCarthy, F. J. Bird, A. L. Addis	Oil burners; oil-fired year-round air conditioning unit; oil burning boilers; oil burning water heaters
Titusville Iron Works Co., Titusville, Pa.	552–553	M. A. Cordner, J. H. Legg, L. K. Hillman, D. E. Penning	Boilers
Toledo Pipe Threading Machine Co., Toledo, O.	493	Clarence A. Popp, F. D. Stuart, C. E. Brokaw, Don Coleman	Pipe tools, cutter, reamers, work benches, power drives and pipe machines
Tork Clock Co., Inc., Mt. Vernon, N. Y.	267	H. deBonneval, B. G. Ferris, C. J. White	Time control apparatus
Torrington Mfg. Co., Torrington, Conn.	207	Alexis Doster, A. W. Evans, W. L. Upson, H. M. Risedorf, A. M. Hennequin, N. E. Doyle	Fan blades and blower wheels
Tuthill Pump Co., Chicago, Ill.	567	W. J. Wagner, H. F. Conner	Pumps for domestic and industrial oil burners
Tuttle & Bailey, Inc., New Britain, Conn.	271–272		Registers and grilles
United States Radiator Corp., Detroit, Mich.	306–307–308–309	E. S. White, R. P. Henderson, J. F. McIntire, W. E. Mosher, R. F. Connell, E. G. Raymond, T. R. Flanagan, D. E. Welsh, E. A. Thompson, W. A. Russell, H. L. Hosking, J. R. Collette	Boilers, radiators, air conditioning units, heating accessories
United States Register Co., Battle Creek, Mich.	25	A. A. Stagg, Harry G. Hollman, J. S. Waggoner, F. C. Bowers, C. J. Pearson	Air conditioning registers, vents, grilles; gravity registers and faces; dampers
United States Steel Corp., Pittsburgh, Pa.	52–53		Sheets; pipe; steel products. Galvanized, galvannealed, stainless
Utica Radiator Corp., Utica, N. Y.	437–438–439–440	Grant H. Browne, Walter A. Sherbrooke, Minor Highes, E. S. Lea, S. W. Shermet, K. J. Walter, C. A. Sawade	Air conditioner; oil boiler; radiators; convectors
Viking Air Conditioning Corp., Cleveland, O.	568	Marion I. Levy, A. Bennekamper, W. E. Van Horne, Harry W. Fiedler, Geo. E. Sumner, Harry Boling	Three-speed conditioner for forced air heating; blower; attic ventilating equipment; warm-air furnace humidifier
Viking Pump Co., Cedar Falls, Ia.	457		Complete line of fuel oil burner pumps
Vilter Mfg. Co., Milwaukee, Wis.	31	F. D. Kirk, H. C. Guild, G. A. Twombly, T. O'Brien, F. T. Flocke, E. E. Schaefer	"Freon" compressors and self contained units
Volcano Burner Corp., New York, N. Y.	564	E. Camerino, W. Camerino, E. Chanvat, June Camerino	Oil burner boiler units
Vortex Mfg. Co., Chicago, Ill.	461		Oil burner
Voss Co., J. H. H., New York, N. Y.	429–430	J. H. H. Voss, A. C. Voss, Karl Schartel, George Gerstmayr, Henry Hoerber, Joseph Weigert, A. Streng	Refrigeration valves
Wagner Electric Corp., St. Louis, Mo.	517–518	A. D. Pool, B. T. McCormick, H. N. Felton, P. A. Shepherd, W. A. Murphy, G. E. Edscorn, G. L. Neeley	Motors for compressors, pumps, water circulators, fans, blowers, unit heaters, stokers; water cooled motors
Ward Machinery Co., Chicago, Ill.	435	W. M. Ward, C. H. Bishop, A. T. Thompson, H. Flagler	Corner lock forming machine; spot welder; edger machine; electric unishear
Waterfilm Boilers, Inc., Jersey City, N. J.	211	A. R. Hanson, P. A. Crammond, E. K. Wodecka	Boiler; air conditioner
Waterloo Register Co., Waterloo, Ia.	421	John C. Knox	Air conditioning grilles

Name of Exhibitor	Booth No.	Representatives in Attendance	Products on Exhibition
Waterman-Waterbury Co., Minneapolis, Minn.	338–339	H. G. Cross, F. G. Sedgwick, M. C. Barnum, Wm. Scott	Oil burning and stoker furnaces; oil burning and stoker winter air conditioner
Watts Regulator Co., Lawrence, Mass.	236		Hot water circulators and specialties
Webster & Co., Warren, Camden, N. J.	43–44B		Heating specialties. See also Anthracite Industries
Webster Electric Co., Racine, Wis.	18		Fuel units, ignition transformers
Weil-McLain Co., Chicago, Ill.	321–322	Thos. D. Casserly, W. R. Stockwell, Jr., Harry Booth	Boilers; radiators
Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.	91–92		Motors, controls, air conditioning equipment
White Manufacturing Co., St. Paul, Minn.	209A	E. H. White, Warren T. White, L. E. Turner, H. W. Clemons	Heat regulators; air and water regulator switches; thermostats and new time clock control
Whiting Corp., Harvey, Ill.	296–297		Stokers
Whitney Metal Tool Co., Rockford, Ill.	479–480	John Jensen, Gus. Jensen	Punches, shears, brakes, angle iron machinery, rolls
Williams Oil-O-Matic Heating Corp., Bloomington, Ill.	225–226	C. U. Williams, W. W. Williams, Earl Nesmith, W. J. Brevitt, Frank Brown, Don Frank, Scotty Muirhead, Earl Ross, R. D. Marshall, John I. Raymond, L. G. Brown, Carl Gentz	Oil burners; water heaters; cooling and year 'round air conditioning units
Wilson & Co., H. A., Newark, N. J.	"A" Mez.		Thermostatic bi-metals
Wing Mfg. Co., L. J., New York, N. Y.	30	A. E. Seelig, H. W. Wheller, J. M. Hancock, C. L. Meyer, T. R. Peyrek, J. H. Carbone, J. Brower, P. J. Wilson, C. H. Smith, G. Nagoshimer, E. H. Hubert	Unit heaters; propeller fans; exhausters; blowers; turbines
Wood Industries, Inc., Gar, Detroit, Mich.	246–247	Gar Wood, Logan Wood, Frank H. Dewey, Donald J. Luty, Gilbert U. Radoye, Daniel M. Kagay, A. F. Beach, Jr., E. R. Parkhurst, C. C. Fox, George Hewitt	Direct-fired air conditioners; boiler-burner units
Worthington Pump & Machinery Corp., Carbondale Div., Harrison, N. J.	413–414	A. H. Baer, T. E. Carpenter, T. J. Atkins, F. J. Watt	Central air conditioners; shower condensers; condensing units; compressors, pumps
"X" Laboratories, Inc., New York, N. Y.	558	E. R. Tolfree, Parke H. Holton, L. D. Patterson, Eugene Norman, Paul Arnold, Edgar Giles, Charles Beckman	Boiler compounds
Yarnall-Waring Co., Philadelphia, Pa.	541–542	B. G. Waring, D. R. Yarnall, H. S. Webster, C. Wilson, J. McKee, J. Kildare, H. C. Woodward	Blow-off valves; water columns; water gages; expansion joints; spray nozzles; steam traps
York Ice Machinery Corp., York, Pa.	50–51	S. E. Lauer, J. R. Hertzler, W. E. Barnum, E. R. Walsh, W. E. Zieber	Economizer; air conditioner; oil burning boiler
Young Radiator Co., Racine, Wis.	275–276	F. M. Young, W. H. Schleck, M. F. May, J. J. Hilt, C. E. Pugh, L. B. Ray, E. R. Clement, C. R. Campion, W. T. Wrightson, E. E. Adams	Unit heaters; convectors; air conditioning units; water coolers; evaporators; heat transfer surfaces
Young Regulator Co., Cleveland, O.	569	E. O. Young, L. C. Ruch, Harry Fiedler	Regulators; split damper control
Youngstown Sheet & Tube Co., Youngstown, O.	17		Pipe and sheets

Data on Power Used for Air Conditioning Given in This Month's Heating ♦ Piping and Air Conditioning Summary Sheets





ANALYSIS BY MONTHS
(Upper Figures are kwhr; Lower Figures are kw)

Establishment	kwhr per Hp per Season	April	May	June	July	Aug.	Sept.	Oct.	Nov.
General Offices	667	486 26	4980 46	5820 54	19100 85	19680 82	12750 79	1293 10	
Doctors' & Dentists'	482		20.2 0.53	23.4 0.50	65.0 0.68	92.0 0.67	98.0 0.67	25.8 0.70	6.0 0.22
Restaurants	912	282 0.7	1130 17.0	3213 24.0	11045 26.0	11975 27.0	6944 26.0	715 8.0	
Beauty Parlors	533		75 1.93	230 2.61	1313 6.12	1251 5.96	618 5.98	132 2.76	21 0.65
Theaters	490		1597 56	2859 61	13374 80	13247 81	5927 75		
Funeral Parlors	310	130 1.6	420 7.8	400 9.6	1190 11.1	1420 11.4	1057 11.0	300 8.5	145 3.2
Department Stores	505		8065 165	22245 237	48585 319	57620 342	44725 285	15190 91	
Candy Stores	911	18 0.3	80 1.1	133 1.5	413 1.6	523 1.6	364 1.5	133 1.3	26 0.8
Drug Stores	1330		175 2.0	760 4.0	1480 5.3	2260 5.7	2030 5.6	960 5.5	
Hotels	782		952 27	3167 40	14195 45	14906 45	6033 43	453 14	
Shoe Stores	675	312 11.4	1878 12.1	2342 12.0	3285 12.9	3249 12.7	1919 12.4	318 11.5	

For complete information, refer to the original article from which these data were obtained, as the purpose of these summaries is not to take the place of the article, but rather to put the essentials into convenient reference form. . . . The Editors will appreciate your suggesting what type of material you'd like presented in this way. . . . Address HEATING, PIPING AND AIR CONDITIONING, 6 North Michigan Avenue, Chicago, Illinois.

Summary Sheets Published to Date

Air Conditioning—Electricity Used by Air Conditioning, by William P. Rock

Amount of electricity used by air conditioning equipment determined from survey of monthly meter readings; table of kilowatthours per horsepower for cooling season; weather conditions during the season; analysis of electrical load by months for general offices, doctors' and dentists' offices, restaurants, theaters, beauty parlors, funeral parlors, department stores, drug stores, shoe stores, and hotels.

Boilers—Selecting the Right Size Heating Boiler, by Sabin Crocker

Factors in selection of right size heating boiler; absorbing capacity of boiler surface; ratings; heat emissions, condensation factors and equivalent ratios for radiating surfaces; conversion factors for heat emitted by column or tube radiation and equivalent nominal radiation; variation with pipe size of heat transfer through insulation and bare pipe; computation of load due to piping; heating power of pipe coils in water storage tanks; heat emission from horizontal pipes; computation of load due to domestic hot water requirements; water heating load vs. temperature rise for various gph rates; warming-up period and allowances; computing the time required for warming-up; tables of weights of pipe and fittings; fuel burning and combustion rates; square feet of boiler heating surface per square foot of grate area; setting heights for stoker fired boilers; furnace volume and furnace height; oil fired boilers; gas fired boilers.

Heat Transfer—How to Solve Heat Transfer Problems, by William Goodman

Heat transfer problems classified; formulas for solving the various problems; illustrative example; tabular data for use with formulas; graphs illustrating determination of large temperature difference and small temperature difference where one fluid is vaporizing, where one fluid is condensing, for parallel flow, and for counterflow.

Pipe—Properties of Pipe, by Arthur McCutchan

Dimensional properties of pipe used in computing velocities, pressure drops, and flexural characteristics for Schedule Numbers 10, 20, 30, 40, 60, 80, 100, 120, 140, and 160 as given in American Standard for Wrought Iron and Wrought Steel Pipe; nominal pipe size, outside diameter, wall thickness, inside diameter, inside diameter squared, inside diameter to fifth power, area of metal, internal cross-sectional area, external surface, moment of inertia, weight of pipe per foot, and weight of water per foot of pipe; significance of the tabulated properties.

Electricity Used by Air Conditioning†

By William P. Rock*

How much electricity is used by air conditioning equipment? Information which is helpful in answering this question was obtained in a survey of monthly meter readings for the year 1936 of installations in Chicago which operated during the entire cooling season. With a knowledge of the price of electric service in different localities, these figures may be readily translated into the cost of operation in cities with comparable climatic conditions.

Each of the classifications analyzed is expressed in terms of the *average installation* for that classification. Some classifications are the result of averaging three or four establishments, and some are the result of averaging as many as fifty. Each is as inclusive as available records would permit.

The tabulation shows numerically the month by month kilowatthour and kilowatt readings on the air conditioning equipment only. The average kilowatthour usage per connected horsepower for air conditioning is also given for each classification. Since one horsepower operating one hour uses very close to one kilowatthour of energy, these figures indicate approximately how many hours, during the season, the average installation has been operated at the equivalent full load.

Some of the classifications had some usage prior to April and after November.

*Air Conditioning Division, Commonwealth Edison Co.

†Summarized from an article published in HEATING, PIPING AND AIR CONDITIONING, April, 1937.

Weather Conditions, Chicago, March-November, 1936

MONTH	AVERAGE TEMP. F	RELATIVE HUMIDITY, PER CENT		
		7 A. M.	NOON	7 P.M.
April	43.6	71	58	64
May	64.4	70	52	56
June	64.6	70	58	59
July	76.8	66	55	58
August	75.0	78	59	67
September	68.2	82	65	73
October	53.6	78	62	68
November	38.0	73	62	66

Heating · Piping
and Air Conditioning

AMERICAN SOCIETY *OF* HEATING *AND* VENTILATING ENGINEERS

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Condensation Within Walls

By F. B. Rowley,* A. B. Algren,** and C. E. Lund*** (*MEMBERS*)
Minneapolis, Minn.

THE condensation of moisture or formation of frost in the various parts of buildings in which low temperatures and high humidities occur has always produced a troublesome problem. It is caused by a combination of conditions, and in cold climates may be expected to a greater or lesser extent in any public building, private home, industrial building such as a creamery or laundry, or any other structure in which high humidities are carried. It may be found in cold storage buildings located in any climate. The problem is not new, but in the past has been accepted as inevitable under certain conditions and has been reasonably well solved by using special construction and other precautions in those places where trouble was apt to occur. Recently there have been marked advances in the science of air conditioning and building construction. Changes have been made which have completely upset the satisfactory application of the old and accepted standards to the newer types of buildings and living conditions. Many of our residences and public buildings which are located in colder climates are now in the danger zone, whereas under former conditions they would have been entirely free from the possibilities of condensation. The magnitude of these new moisture problems has caused some people to doubt the real value of the improvements. Others who accept the benefits of the improved conditions are inclined to shift the responsibility for any difficulties encountered to one of the several causes in which they are not specifically interested. Thus to some the entire trouble is due to air conditioning, to others it is insulation, and to still others it is due to the application of weatherstrips and other methods of providing tighter building construction.

The condensation of moisture takes place when water vapor is cooled below its saturation temperature. In the cases under consideration air is also present with the vapor, but the presence of air does not change the condensing temperatures of the vapor. This condensation of moisture may occur on the surface of a wall or at any place within the structure where the temperature of the material is below the saturation or dew-point temperature of the vapor. If these temperatures are below the freezing point then the condensation will result in an accumulation of frost and ice. The temperatures of the surfaces and interior parts of the wall may be calculated with reasonable accuracy from the known thermal properties of the materials used in the walls. The dew-point temperature of the air to be carried within a room may be obtained from any psychrometric chart and thus the danger point of any interior wall surfaces may be anticipated. The dew-point temperature of the air within the walls may also be determined from a psychrometric chart providing its conditions are known. Unfortunately the vapor transmitting properties of building materials are not so well understood as the heat transmitting properties are and the moisture conditions within a wall cannot be calculated so readily. An extended study of this phase of the problem is under investigation at the Forest Products Laboratory, Madison, Wis., and a recently published paper[1] contains some valuable information and recommendations concerning the moisture problem. The author has made an excellent analysis of the problem and has shown by diagrams how insulation will reduce the temperatures of the outside sections of a wall and may thereby increase the possibilities of trouble in these areas.

The fundamental causes of condensation problems are high humidities and low temperatures of the materials with which the vapor comes in contact. Several of the changes which have recently taken place in air conditioning requirements and building construction have greatly increased the probability of meeting these conditions. These changes are: first, raising the relative humidity within the structure by artificial means, second, the application of weatherstrips and other means to prevent the filtration of air through the walls of the building, and third, the addition of better insulation in the walls of the building. These three factors are all necessary if the benefits of modern scientific developments are to be enjoyed and all have contributed to the moisture problem. It cannot be solved by placing the responsibility on any single factor.

For many years the low relative humidity found in the average residence and public building located in cold climates has been a subject of much discussion and the effect of these low humidities on both health and furnishings has been controversial. The net result has been an increase in the average relative humidities carried. Even though these higher relative humidities are often under control, they have contributed largely to the problem of condensation and frost formation within walls.

In recent years the loss of heat from a building due to the leakage of air around windows and doors has become more fully appreciated and various types of weatherstrips have been extensively used to reduce this exchange of air between the inside and outside of the building. The absolute humidity of the outside air is usually much lower than that of the inside air and the natural result has been to build up the moisture content of the inside air by the normal processes within the building even though artificial humidification is not used.

Increasing the relative humidity of the air in a building or adding weatherstrips to a building will increase the probability of condensation troubles, but the addition of insulation to the walls may either increase or decrease this probability, depending upon the type of construction and the part of the wall under consideration. Since insulation resists the passage of heat, that portion of the wall on the warm side of the insulation will become warmer and that portion on the cold side will become

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[1]Condensation in Walls and Attics, by L. D. Teesdale (*U. S. Forest Products Bulletin*).

colder after the insulation is applied. If the inside wall surfaces are being considered the conditions are improved. If, however, consideration is being given to that portion of the wall on the cold side of the insulation, conditions may be aggravated due to the lower temperatures, especially if the moisture can penetrate to this part of the wall.

In many instances there is no logical argument against the use of higher relative humidities for better conditions of comfort and health, nor against the addition of weatherstrips and other means of reducing the air leakage between the inside and outside of a building. Much less could any argument be applied against the addition of insulation. Each of these are essential factors for a well constructed building and a comfortable place in which to live. Each factor may be considered as contributing to the condensation problem, but the elimination of any one would not be a positive cure. The problem must be solved by using a type of construction which will permit the use of all three factors and prevent condensation under all reasonable conditions.

Vapor must be considered as a material by itself independent of the air with which it is mixed. It has the same density, pressure, and dew point temperature regardless of whether air is present or not. It moves from one place to another without the necessity of moving the air. It passes through some materials very readily while others offer a high resistance to its passage. The amount passing through any material is considered proportional to the drop in vapor pressure along the path of flow through that material, although this law should have further investigation. The following are some of the factors which must be considered in making an analysis of a particular problem: first, the air conditions to be maintained within a building with particular reference to its dry-bulb and dew-point temperatures; second, the outside weather conditions to be expected with reference to wind velocity, minimum temperatures, and duration of low temperature periods; third, the conductivity of the wall under consideration, with particular reference to the probable temperatures of its inner surface and interior sections; and fourth, the vapor proofness of the materials from which the wall is constructed, with particular reference to its inner and outer surface linings.

It is usually not difficult to estimate reasonable inside air temperatures and humidities which are to be carried in the building, but if these requirements are not adhered to in practice they may be directly responsible for condensation difficulties. Since outside weather conditions cannot be controlled an intelligent study of Weather Bureau records should be made to find what extreme conditions may be expected for any particular location. The thermal conductivities of most materials and combinations of materials are sufficiently well understood to make calculations for the wall temperatures to be expected under maximum design conditions. The vapor transmitting properties of various materials under different conditions have not as yet been well defined and this presents one of the most difficult steps in designing a structure which will be free from condensation troubles. It is easy to determine the dew-point temperature of the air which is to be maintained within a building and to calculate the minimum outside weather conditions which may be met without lowering the inside surface temperature below the dew-point temperature of the air, but it is not easy to calculate the dew-point temperature which may be expected for the vapor in the interior parts of the wall. Vapor travels from a point of high vapor pressure to a point of lower vapor pressure, which in the case of ordinary walls in cold climates is from the inside surface to the outside surface of the wall. If the inner surface has a low resistance to the passage of vapor and the outer surface has a high resistance, then in extreme cases the interior part of the wall may have vapor pressures equal to the air inside of the building. If, on the other hand, the inside surface has a high vapor resistance and the outside a low vapor resistance, then the vapor in the interior parts of the wall may have a dew-point temperature more nearly equal to that of the outside air. Since condensation of moisture can only take place when the wall temperatures are below the dew-point temperature of the vapors present, the logical solution is to select materials for the inside sections of a wall which have a high vapor resistance, and for the outside sections materials which have a low vapor resistance. In so far as possible the vapor should be stopped on the high temperature side of the wall and not on the low temperature side. In most cases the vapor density within any section of a wall will be governed by the difference in vapor pressure on the two sides of the wall and the vapor resistance of the materials on each side of the section under consideration. By the proper selection and application of materials in a wall the rate of vapor travel through the various parts of the wall may be controlled, and therefore the vapor pressure within the wall may be regulated and maintained at a safe level for any temperature conditions which may be expected within the wall.

Recognizing that there are condensation problems with many types of building construction in combination with various air temperatures and relative humidities as found in modern types of construction, an extensive research program has been initiated to investigate the several factors. The aim is first, to determine the extent of the condensation problem under normal and extreme conditions; second, to obtain a measure of the relative value of the various methods of correcting these difficulties; and third, to find the limiting conditions under which various types of building construction may be used without excessive condensation troubles and still retain the benefits of higher humidities and good building construction.

It was decided to approach the problem in a practical way and to make tests which would apply to full scale building construction under temperature and humidity conditions which might be expected in the severe climates. To accomplish this a test room was constructed 30 ft square and 25 ft high in which it was possible to control the air temperatures to a minimum of 30 F below zero. Small size test houses were constructed and placed within this room with provisions for maintaining the air temperature and humidities within the houses at any desired level. The houses were so constructed that they might be taken apart and examined from time to time throughout a test period to determine the physical conditions of the interior of the walls and to make comparisons between various types of structures. After fundamental data are obtained covering a sufficient variety of typical construction, larger scale models are to be constructed to prove out on them the value of the

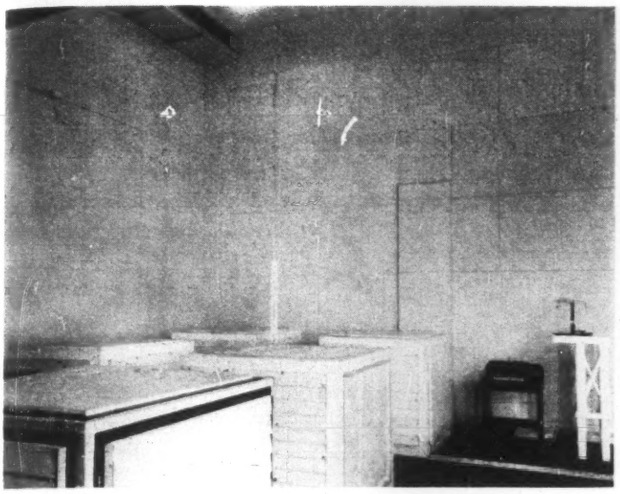


Fig. 1—View of cold room showing test house in place and weighing balances

more promising designs. The test apparatus has been completed and tests on the small size models have been in progress for approximately two months; the results of these tests will be presented in this discussion.

Test Apparatus

The test room is shown in the photograph of Fig. 1 and the line drawing of Fig. 2. The set-up for tests with six small houses in place is shown in Fig. 1. A vertical cross-sectional view drawn through the center of the room looking toward the front is illustrated in Fig. 2. The room is 30 ft square and 25 ft high and is insulated throughout with 6 in. of rock wool. It is constructed with 2x6 studs lined on the inside surface with ½ in. insulating board and no interior finish. The outside surface over the studs is thoroughly sealed with a vapor resisting paper cemented at the joints and finished on the outside with ½ in. insulating board. The 6 in. floor joists are sealed on top and bottom with the vapor proof paper and the space between is filled with rock wool. The object in constructing the room was to prevent the leakage of any outside moist air into the insulation, and

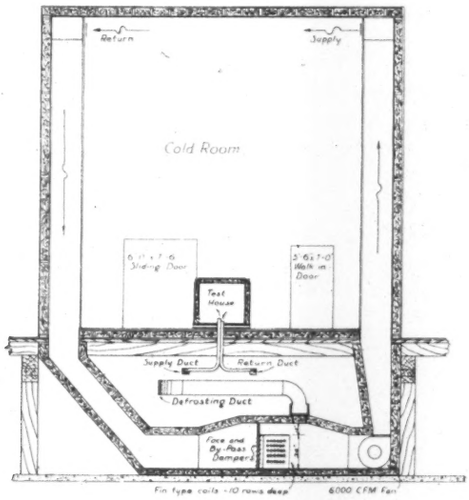


Fig. 2—Sectional view through cold room and cooling unit

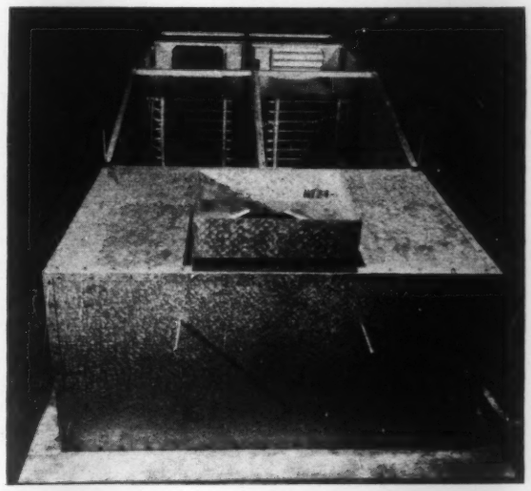


Fig. 3—View of cooling unit while under construction

to allow a slight amount of breathing through the cold surface. Thus far this type of construction has proved satisfactory and it has been possible to hold the temperature in the room to 25 F below zero. Two doors are provided, one of them 6 ft x 7 ft 6 in. used to move large pieces of equipment into and out of the room, and the other a small service door 3 ft 6 in. x 7 ft 0 in. The room is provided with nine 10 in. square openings in the floor through which conditioned air may be supplied to each test unit.

The room is cooled by a 25 ton ammonia compressor with direct expansion fin type cooling coils. The cooling unit is placed in the basement under the test room and air is circulated through the cooling unit up through a vertical duct in the right hand corner of the room, Fig. 2, and distributed from a horizontal duct at the ceiling. Air is collected by a similar duct at the ceiling on the opposite side of the room and returned to the cooling unit. A sectional view of the cooling unit is shown below the test room in Fig. 2, a top view from the air discharge end during the period of construction is shown in Fig. 3, and a completed view is shown in the foreground of the photograph of Fig. 4. Air is circulated through the cooling unit by a 6000 cfm fan located on the discharge side of the unit. The air is drawn through the cooling unit in two parallel streams passing through identically constructed cooling compartments either of

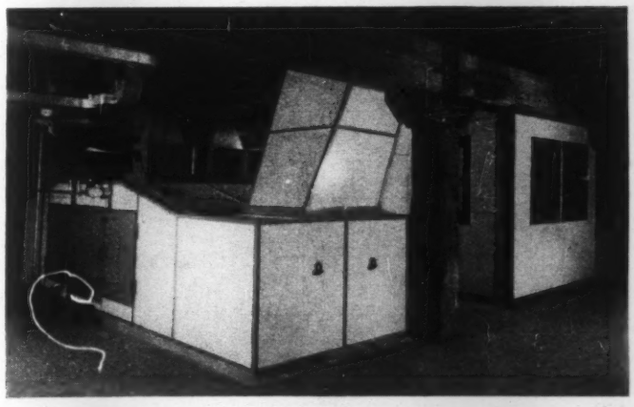


Fig. 4—View of completed cooling unit and one end of control room

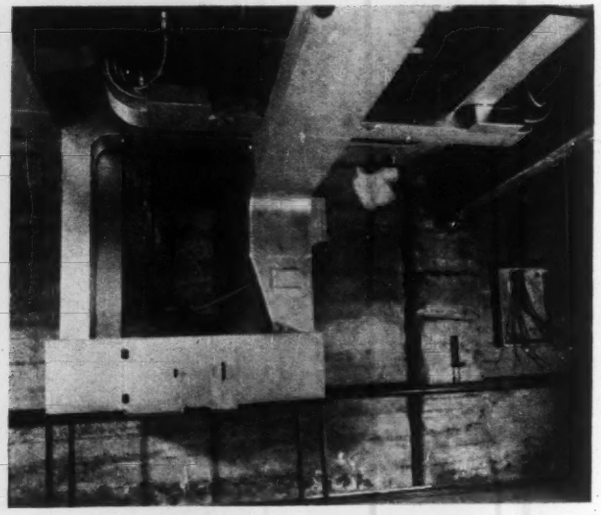


Fig. 5—Air conditioning unit for test houses

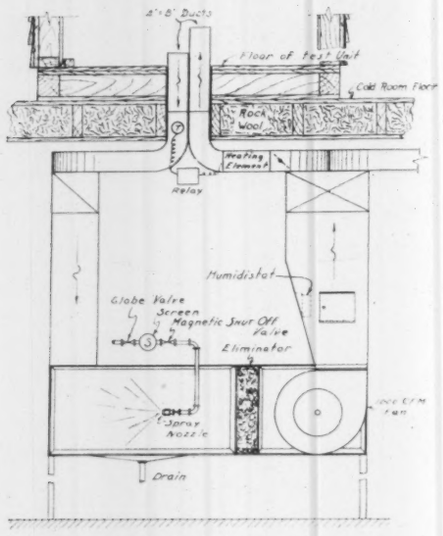


Fig. 6—Sectional view of air conditioning unit with distribution ducts to test houses in cold room

Fig. 7—Control and thermocouple panel for test equipment

which may be completely closed off and connected with recirculated room air for defrosting purposes. A shut-off door is provided at each end of each compartment and one compartment may be operated independently of the other. Each air passage is provided with a face and by-pass damper at the entrance to a direct expansion ammonia cooling coil. The cooling coil is 10 rows deep, 18 in. wide by 34.5 in. high with a by-pass damper over the top of coils 18 in. wide and 12 in. high. The complete unit is enclosed in a galvanized iron case covered on the outside with 6 in. of rock wool and ½ in. insulating board.

A defrosting fan delivers air at room temperature through either of two ducts shown at the top of the unit in Fig. 4. One of these ducts is connected to each cooling compartment and dampers are so arranged that when this cooling compartment is shut off from the main circulating fan the room air may be circulated through the cooling coils and discharged out into the room for defrosting purposes. In practice it has been found necessary to defrost once in two days and a period of 15 min is sufficient for defrosting each side of the equipment.

Conditioned air is supplied to each of the test units in the large cold room through openings in the floor, which are directly under each test unit, from an air conditioning plant located in the basement and shown in the photograph, Fig. 5, and the line drawing, Fig. 6. The conditioning plant controls the humidity and circulates the air through branch ducts distributing it through the floor openings into the small test houses. The air from the test houses is collected and returned to the air conditioning unit for reconditioning. The temperature of the air is controlled independently for each test house by electric heating elements placed in the supply duct and controlled by a thermostat placed in the return air line for the particular unit. The electrical heaters are arranged so that a part may be connected to a constant supply and a part on a thermostatic control line. Thus the load may be controlled with very little fluctuation in the supply line.

All of the control instruments and thermocouples are brought to a control panel shown in the photograph of Fig. 7 and located in the instrument room which is shown at the right on Fig. 4. At the top of the control panel there are nine switches and pilot lights for the heating elements of the nine test sections located in the cold room. At the right central section is shown the switch and pilot light of the humidity control for the air conditioning unit, and at the central and left central section of the panel are shown the controls for the motor which operates the face and by-pass dampers in the cooling unit to control the temperature in the cold room. At the lower section of the board there are switches for 60 thermocouples which are wired directly to corresponding panels on the inside walls of the large cold room. Thus thermocouples may be connected directly from the test houses to the panels within the cold room and the readings taken at the control board without the necessity of running wires from the cold room to the control board for each set-up.

Fig. 8 shows at the left a view of the 25 ton ammonia compressor used for the cooling units, at the central sections a view of the control instrument room, and at the right a view of the defrosting fan and discharge

ducts. These are located in the basement underneath the large test room.

Test Houses

The test houses were all of frame construction and 4 ft x 4 ft x 4 ft inside dimensions. The main frame for supporting the walls and ceiling of each house was built up with 4 in. x 4 in. corner posts attached rigidly to the four corners of the floor and connected at the top with a 4 in. x 4 in. frame. The walls were constructed in sections so that they could be inspected for frost formation or other changes in conditions during test periods. Fig. 9 shows a photographic view of two of the frames located in position for test in the cold room with one of the walls applied. Fig. 10 is an oblique sectional drawing of a complete house showing in detail the methods of applying walls to the main frame, and of sealing the interior and exterior joints. Fig. 11 shows an oblique sectional drawing of one of the walls with the sheathing and siding removed as provided for in the method of construction.

The walls were constructed of 2 in. x 4 in. studs spaced 16 in. on center with a 2 in. x 4 in. plate at top and bottom. In all cases the interior finish was made up of metal lath and three coats of plaster; a scratch, brown and finish coat, making a total of ¾ in. in thickness. The outside sections of the walls were constructed with 25/32 in. pine sheathing, building paper, and 6 in. redwood siding. For Walls Nos. 1 to 16 inclusive, 6 in. pine butt sheathing was used, and for all other walls 8 in. pine shiplap was used. The outside construction was nailed to 1 in. square nailing strips which were fitted into corresponding grooves in the outside surface of the studs. The assembled outside unit was securely applied to the studs using wood screws instead of nails so that it could be removed for inspection at various periods of the test. The assembled wall was provided on the outside with a 2 in. x 2 in. x ⅛ in. angle iron frame as shown in Fig. 11. When the assembled wall was applied to the test house the angle iron frame was securely fastened and sealed to the 4 in. x 4 in. framework.

The ceilings were constructed with 2 in. x 4 in. joists spaced 16 in. on center with metal lath and 3 coats of plaster on the inner surface, and with building paper and 6 in. pine butt sheathing on the outer surface. The outer surface finish was nailed to 1 in. square nailing strips and fastened to the studs with wood screws in the same manner as for the side wall construction. A 2 in. x 2 in. x ⅛ in. angle iron frame was applied around the outside of the finished ceiling. The floors of the test unit were built with 2 in. x 4 in. joists spaced 16 in. on center with building paper on top and bottom and covered on the lower side with 6 in. butt sheathing, and on the upper side with rough flooring and a finished fir floor. The floors were made integral with the 4 in. x 4 in. framework and not removed or opened for inspection during the test.

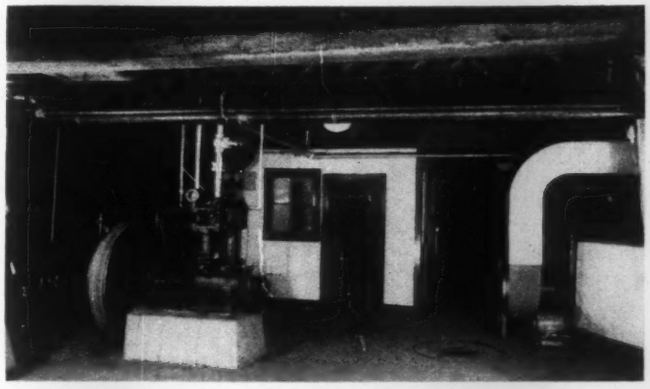


Fig. 8—Ammonia compressor, control room and defrosting fan for cooling equipment

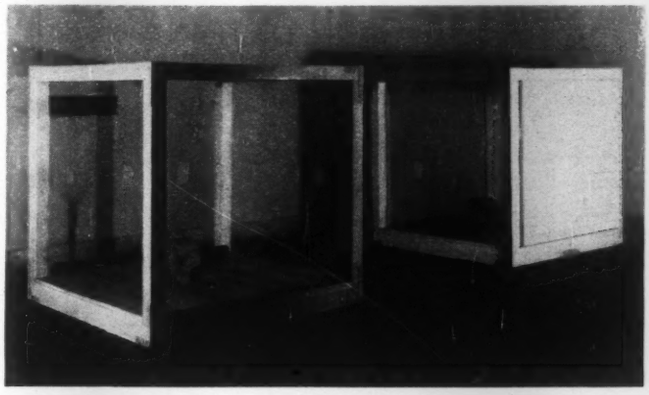


Fig. 9—Frames for test houses set in cold room with one wall applied

The first and second coats of plaster were made of beaver fibered plaster and the finished coat was ivory hydrated finishing lime. In all cases the plaster was allowed to dry at least 14 days before making tests. The paint used for the outside surface of the walls was made up of 70 per cent pigment and 30 per cent vehicle by weight. The pigment was a pure white lead paste and the vehicle was 90 per cent linseed oil and 10 per cent dryer.

As indicated in Table 1 showing wall construction, some of the walls were insulated

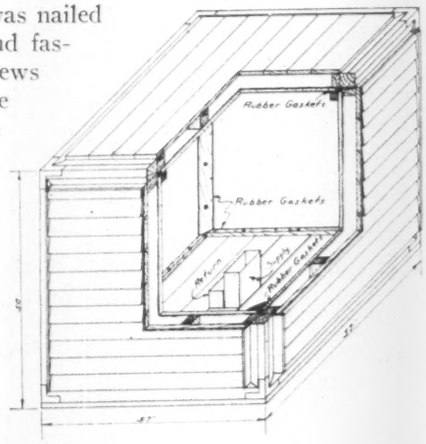


Fig. 10—Sectional view of test houses showing details of construction

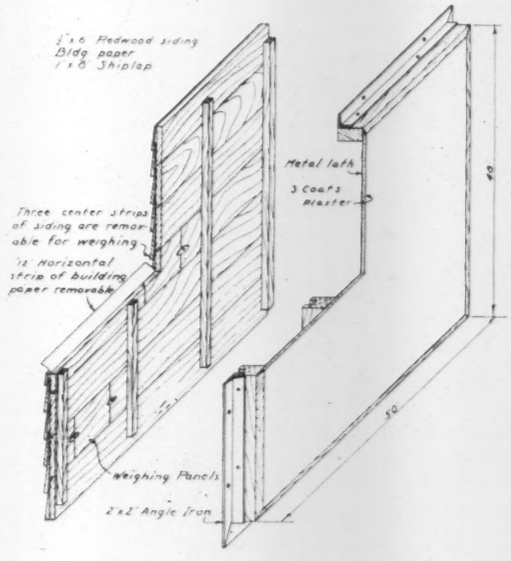


Fig. 11 (*right*)—Type of wall construction used to facilitate inspection

and some were not. Vapor proof barriers were added to the inside surface of part of the walls in the form of a surface paint or in the form of a vapor proof paper placed between the metal lath and studs. The outside sections of some of the walls were changed by omitting the building paper and by drilling ventilating holes through the sheathing.

When a test unit was assembled by applying the walls and ceiling to the main frame, the joints between the angle iron frame around each wall and the main frame of the test unit were sealed with putty, and the joints along the interior corners were sealed with sealing strips and rubber gaskets to prevent the infiltration of air. The test house was set in the cold room and thoroughly sealed to the floor to prevent air leakage under the floor of the test house. The conditioned air was circulated through the test house by the ducts as shown in Fig. 10.

For each wall three sections of the sheathing, one board in width and 8 in. long were cut out during the construction, closely fitted into place and applied with turn buttons. The loose sections were removed and weighed at periods during the tests to measure the amount of moisture and frost accumulated on the sheathing.

Table 1—Details of Wall Construction[a]

Wall No.	Test No.	Vapor Barrier: Inside Surface of Plaster	Vapor Barrier: Between Metal Lath and Studs	Insulation Between Studs	Aluminum Panel on Sheathing Between Studs	Notes
1	1	No	No	No	No	
1-A	2,3	2-coats white flat paint	No	3⅝ in. rock wool	[b]Yes	
2	1, 2, 3	No	No	No	Tests 2, 3 only	8-thermocouples in wall
3	1	No	No	No	No	
3-A	2, 3	2-coats aluminum paint	No	3⅝ in. rock wool	[b]Tests 2, 3 only	
4	1	No	No	No	No	
4-A	2, 3	1-coat size; 2-coats white flat paint	No	3⅝ in. rock wool	[b]Yes	
5	1	No	No	3⅝ in. rock wool	No	2-¼ in. holes drilled in top and bottom of outer stud section during test
5-A	2	No	No	3⅝ in. rock wool	Yes	2-¼ in. holes in top and bottom of each stud section
5-B	3	No	No	3⅝ in. rock wool	Yes	2-1 in. holes in top and bottom of each stud section
6	1, 2, 3	No	No	3⅝ in. rock wool	Tests 2, 3 only	8-thermocouples in wall
7	1	No	No	3⅝ in. rock wool	No	2-⅛ in. holes drilled in top and bottom of outer stud section during test
7-A	2, 3	2-coats size	No	3⅝ in. rock wool	[b]Yes	
8	1	No	No	3⅝ in. rock wool	No	
8-A	2	1-coat size; 2 coats aluminum paint	No	3⅝ in. rock wool	[b]Yes	
9	1, 2, 3	No	A	3⅝ in. rock wool	Tests 2, 3 only	8-thermocouples in wall, paper lapped at joints
10	1, 2, 3	No	A	3⅝ in. rock wool	Test 3 only	Paper lapped at joints
11	1, 2, 3	No	B	3⅝ in. rock wool	[b]Tests 2, 3 only	8-thermocouples in wall. Paper lapped at joints
12	1, 2, 3	No	B	3⅝ in. rock wool	Test 3 only	Paper lapped at joints
13	1, 2, 3	No	A	3⅝ in. rock wool	[b]Tests 2, 3 only	8-thermocouples in wall. Paper sealed at joints, with asphalt
14	1, 2, 3	No	A	3⅝ in. rock wool	Test 3 only	Paper sealed at joints with asphalt
15	1, 2, 3	No	B	3⅝ in. rock wool	Test 3 only	Paper sealed at joints with asphalt
16	1, 2, 3	No	B	3⅝ in. rock wool	Tests 2, 3 only	Paper sealed at joints with asphalt
17	3	No	Bldg. Paper	3⅝ in. rock wool	Yes	
18	3	No	Bldg. Paper	3⅝ in. rock wool	Yes	
19	3	No	No	3⅝ in. rock wool	Yes	
20	3	No	No	3⅝ in. rock wool	Yes	No bldg. paper between sheathing and siding
21	3	No	No	3⅝ in. rock wool	Yes	One 2 in. hole at top of each stud section
24	3	1-coat asphalt	No	3⅝ in. rock wool	Yes	

[a]All walls were built with 2 in. x 4 in. studs spaced 16 in. O. C., metal lath and three coats of gypsum plaster on inside surface, 25/32 in. pine sheathing, 50 lb building paper, 6 in. redwood siding with three coats white lead paint on outside surface. Six inch pine butt sheathing was used in walls No. 1 to 16, inclusive, and 8 in. pine shiplap sheathing was used in all other walls. Vapor barriers were added to the inside section, insulation was added between the studs, and paper was removed from or holes were provided in outside section as indicated for the various walls. In part of the walls aluminum panels were placed against the inside surface of the sheathing between the studs as a means of collecting and weighing condensate. Thermocouples were placed between the aluminum panel and sheathing.
[b]Aluminum panels changed from outer stud section to center stud section between tests 2 and 3.
A = 30, 30, 30 duplex paper, two sheets of 30 lb paper cemented together with 30 lb asphalt.
B = Asphalt impregnated and surface coated sheathing paper, glossy surfaced and weighing 50 lb per roll of 500 sq ft.

Table 2—Details of Ceiling Construction[a]

Ceiling No.	Test No.	Vapor Barrier: Inside Surface of Plaster	Vapor Barrier: Between Metal Lath and Studs	Insulation Between Joists	Aluminum Panel on Sheathing Between Joists	Notes
C-1	1, 2	No	No	No	No	
C-1-A	3	No	No	3⅝ in. expanded aluminum-magnesium silicate	Yes	
C-2	1, 2, 3	No	No	3⅝ in. rock wool	[b]Tests 2, 3 only	
C-3	1, 2, 3	No	B	3⅝ in. rock wool	[b]Tests 2, 3 only	Paper lapped at joints. 5-thermocouples in ceiling
C-4	1	No	B	3⅝ in. rock wool	No	Paper sealed at joints with asphalt
C-4	2, 3	No	B	3⅝ in. rock wool	Yes	Paper sealed at joints with asphalt
C-5	3	No	No	3⅝ in. redwood bark fiber	Yes	
C-6	3	No	No	3⅝ in. sawdust	Yes	

[a]All ceilings were built with 2 in. x 4 in. joists spaced 16 in. O. C., metal lath and three coats of gypsum plaster on inside surface, 50 lb building paper between joists and sheathing, no siding. Sheathing painted with three coats of white lead paint on outside surface. 6 in. pine butt sheathing was used in ceilings 1 to 4 inclusive, and 8 in. pine shiplap sheathing was used in all other ceilings. Vapor barriers were added to the inside section and insulation was added between the joists as indicated. Aluminum panels supplied as described in Table 1.
[b]Aluminum panels changed from outer joist section to center joist section between tests 2 and 3.
B = Asphalt impregnated and surface coated sheathing paper, glossy surfaced and weighing 50 lb per roll of 500 sq ft.

After the first series of tests there was some question as to whether or not some of the vapor which passed through the inner section of the wall was also leaving the walls through the exterior finish. In order to reduce this possibility sheets of aluminum were placed between the studs and against the sheathing, one such sheet being used for each wall. Thermocouples were placed throughout part of the walls and in each case a thermocouple was placed between the aluminum sheet and the wood sheathing in close contact with the aluminum sheet. The temperature indicated by this thermocouple was taken as the temperature of the aluminum sheet.

In a few of the insulated walls a central section of the siding and building paper was so applied that it could be removed for inspection during a test.

Test Procedure

In making a test the test houses were completely assembled, the cooling equipment started, and the room

temperature brought down to test requirements. Test conditions were then maintained for a reasonable time to establish uniform temperature and humidity gradients through the walls. After this preliminary period the walls were inspected to determine their condition without allowing any unnecessary changes in temperature or moisture conditions surrounding them. After these records were made the walls were again closed and the test continued for a period of from 5 to 19 days, after which the walls were again taken apart and inspected to get the rate of frost formation or other changes under constant temperature and humidity conditions. At the end of the test the walls were completely taken apart and all observations made including photographs of typical conditions. The following tests have been run and results given in Tables 3, 4 and 5.

Test 1, first period,	Oct. 6	to Oct. 11	100 hours
Test 1, second period,	Oct. 11	to Oct. 27	360 hours
Test 1, third period,	Oct. 27	to Nov. 2	120 hours
Test 2, first period,	Nov. 6	to Nov. 12	136 hours
Test 2, second period,	Nov. 12	to Nov. 17	114 hours
Test 3, first period,	Nov. 19	to Nov. 22	80 hours
Test 3, second period,	Nov. 22	to Nov. 27	120 hours

As previously stated one of the objects of these tests was to demonstrate some of the causes of condensation in walls, the extent to which it is likely to occur under severe conditions, and some practical methods of overcoming the difficulty without foregoing the comfort and benefits of either air conditioning or good building construction. The tests have been made under conditions which were severe enough and for a sufficient length of time to show the nature of the condensation problems

Table 3—Test No. 1—Data and Results

Wall No.	Photograph Taken Fig. No.	Cold Room F	Test House D. B. Temp. F	Test House Per Cent R. H.	Wall Temperature F: Plaster Inside Surface	Plaster Outside Surface	Air Space Top	Air Space Center	Air Space Bottom	Sheathing Inside Surface	Sheathing Outside Surface	Outside Surface of Siding	Per Cent Change in Weight of Insulation	Moisture Gain Grams per Sq Ft per 24 Hours: Sheathing	Bldg. Paper	Siding
Test Period No. 1—100 Hours																
1		—19 2	70 3	41	—	—	—	—	—	—	—	—	—	—	—	—
2		—19 2	70 3	41	62 0	55 6	50 8	46 2	33 1	38 5	16 2	— 2 0	—	—	—	—
3		—19 2	70 3	41	—	—	—	—	—	—	—	—	—	—	—	—
4	12	—19 2	70 3	41	—	—	—	—	—	—	—	—	—	1 29	—	—
C-1		—19 2	70 3	41	—	—	—	—	—	—	—	—	—	—0 11	—	—
5		—19 2	70 5	41	—	—	—	—	—	—	—	—	—	—	—	—
6		—19 2	70 5	41	68 1	65 7	27 3	26 8	22 8	1 6	— 5 9	—11 7	—	—	—	—
7		—19 2	70 5	41	—	—	—	—	—	—	—	—	—	—	—	—
8		—19 2	70 5	41	—	—	—	—	—	—	—	—	— 0 99	4 07	—	—
C-2		—19 2	70 5	41	—	—	—	—	—	—	—	—	—	—0 23	—	—
9		—19 2	70 7	40	65 1	62 3	25 0	20 8	21 6	— 3 6	— 9 9	—16 0	—	—	—	—
10		—19 2	70 7	40	—	—	—	—	—	—	—	—	—	1 64	—	—
11		—19 2	70 7	40	66 3	61 6	24 6	22 6	15 4	— 3 6	— 9 9	—17 2	—	—	—	—
12		—19 2	70 7	40	—	—	—	—	—	—	—	—	—	0 35	—	—
C-3		—19 2	70 7	40	71 4	67 2	—	24 7	—	— 8 1	—	—13 4	—	—	—	—
13		—19 2	70 3	40	67 9	64 0	23 2	19 6	9 9	— 1 6	— 7 6	—15 9	—	—	—	—
14		—19 2	70 3	40	—	—	—	—	—	—	—	—	—	0 58	—	—
15		—19 2	70 3	40	—	—	—	—	—	—	—	—	—	—	—	—
16		—19 2	70 3	40	—	—	—	—	—	—	—	—	—	0 23	—	—
C-4		—19 2	70 3	49	—	—	—	—	—	—	—	—	—	1 29	—	—
Test Period No. 2—360 Hours																
1		—15 2	70 0	44	—	—	—	—	—	—	—	—	—	—	—	—
2		—15 2	70 0	44	62 0	55 7	50 9	47 1	34 5	38 9	19 2	2 7	—	—	—	—
3		—15 2	70 0	44	—	—	—	—	—	—	—	—	—	0 92	—	—
4[a]		—15 2	70 0	44	—	—	—	—	—	—	—	—	—	0 89	0.125	0 13
C-1[a]	25	—15 2	70 0	44	—	—	—	—	—	—	—	—	—	0 44	—	—
5	13	—15 2	70 0	44	—	—	—	—	—	—	—	—	—	—	—	—
6		—15 2	70 0	44	67 9	65 2	28 6	28 8	26 6	5 5	— 1 0	— 6 3	—	—	—	—
7		—15 2	70 0	44	—	—	—	—	—	—	—	—	—	—	—	—
8		—15 2	70 0	44	—	—	—	—	—	—	—	—	—1 81	2 74	0 010	0 015
C-2[a]	26	—15 2	70 0	44	—	—	—	—	—	—	—	—	—	0 23	—	—
9		—15 2	70 1	43	65 4	62 5	25 5	23 3	23 3	— 0 1	— 6 1	—11 7	—	—	—	—
10[a]	14	—15 2	70 1	43	—	—	—	—	—	—	—	—	—0 53	1 27	0 010	0 010
11		—15 2	70 1	43	66 4	61 9	25 8	25 2	19 3	0 6	— 5 3	—11 9	—	—	—	—
12[a]	15	—15 2	70 1	43	—	—	—	—	—	—	—	—	—0 67	0 96	0 005	0 016
C-3		—15 2	70 1	43	71 3	67 2	—	26 8	—	— 4 0	—	— 8 8	—	—	—	—
13		—15 2	70 3	45	67 7	63 9	24 1	21 8	13 4	2 2	— 3 5	—11 1	—	—	—	—
14		—15 2	70 3	45	—	—	—	—	—	—	—	—	—0 39	0 53	0 010	0 016
15		—15 2	70 3	45	—	—	—	—	—	—	—	—	—	—	—	—
16[a]	16	—15 2	70 3	45	—	—	—	—	—	—	—	—	—0 19	0 39	0 005	0 021
C-4[a]	27	—15 2	70 3	45	—	—	—	—	—	—	—	—	—	0 44	—	—
Test Period No. 3—120 Hours																
1		— 9 9	70	46	—	—	—	—	—	—	—	—	—	—	—	—
2		— 9 9	70	46	62 0	56 3	51 7	48 1	36 1	40 9	21 6	5 2	—	—	—	—
3		— 9 9	70	46	—	—	—	—	—	—	—	—	—	0 85	0 145	0 09
4[a]		— 9 9	70	46	—	—	—	—	—	—	—	—	—	0 83	0 25	0 14
C-1[a]		— 9 9	70	46	—	—	—	—	—	—	—	—	—	0 38	—	—
5[a]		— 9 9	70 4	45	—	—	—	—	—	—	—	—	—6 24	4 25	0 012	0 025
6		— 9 9	70 4	45	68 4	65 8	31 6	30 5	30 0	8 3	2 5	— 2 5	—	—	—	—
7		— 9 9	70 4	45	—	—	—	—	—	—	—	—	—	—	—	—
8		— 9 9	70 4	45	—	—	—	—	—	—	—	—	—2 44	2 72	0 017	0 021
C-2[a]		— 9 9	70 4	45	—	—	—	—	—	—	—	—	—	0 24	—	—
9		— 9 9	69 9	47	65 3	62 6	27 4	25 4	25 1	3 6	— 1 9	— 7 0	—0 12	0 64	0 008	
10[a]		— 9 9	69 9	47	—	—	—	—	—	—	—	—	—1 16	1 03	0 008	0 021
11		— 9 9	69 9	47	66 3	62 1	26 4	27 8	22 4	4 3	— 1 2	— 7 2	—0 03	0 34	0 004	0 012
12[a]		— 9 9	69 9	47	—	—	—	—	—	—	—	—	—0 80	0 63	0 004	0 012
C-3		— 9 9	69 9	47	71 4	67 3	—	29 2	—	0 5	—	— 4 1	—	—	—	—
13		— 9 9	70 2	45	68 0	64 4	26 1	24 4	17 0	5 9	0 5	— 6 5	—0 83	0 34	0 004	0 025
14		— 9 9	70 2	45	—	—	—	—	—	—	—	—	—0 71	0 56	0 008	0 012
15		— 9 9	70 2	45	—	—	—	—	—	—	—	—	—0 86	0 30	0 004	0 021
16[a]		— 9 9	70 2	45	—	—	—	—	—	—	—	—	—0 67	0 42	0 004	0 008
C-4[a]		— 9 9	70 2	45	—	—	—	—	—	—	—	—	—	0 36	—	—

[a]Photographs taken at end of test period No. 2.

and to demonstrate that there are practical methods of meeting these problems. The methods of measuring the extent of the vapor travel through various types of wall construction have been relative and not reduced to an absolute basis. In the first part of the tests the rate of the vapor travel was measured by observation and weight of the frost formed on the inside surface of the sheathing for the different walls. For the last test it was measured by placing an aluminum condensation panel between the studs in contact with the inside surface of the wood sheathing. The test results, however, showed a reasonable check between the amount of mois-

Table 4—Test No. 2—Data and Results

Wall No.	Cold Room Temp. F	Test House		Wall Temperatures F							Outside Surface of Siding	Per Cent Change in Weight of Insulation 250 Hours	Moisture Gain Grams per Sq Ft per 21 Hours		
				Plaster		Air Space			Sheathing						
		Dry-Bulb Temp. F	Per Cent R. H.	Inside Surface	Outside Surface	Top	Center	Bottom	Inside Surface	Outside Surface			Sheathing 250 Hours	Aluminum Panels 250 Hours	Aluminum Panels by Periods
Test Period No. 1—136 Hours															
1-A	— 8 9	69 8	43	—	—	—	—	—	—	—	—	—	—	—	1 99
2	— 8 9	69 8	43	62 9	57 0	52 5	48 9	36 6	41 5	21 8	6 8	—	—	—	0 19
3-A	— 8 9	69 8	43	—	—	—	—	—	—	—	—	—	—	—	1 42
4-A	— 8 9	69 8	43	—	—	—	—	—	—	—	—	—	—	—	1 59
C-1	— 8 9	69 8	43	—	—	—	—	—	—	—	—	—	—	—	—
5-A	— 8 9	70 1	43	—	—	—	—	—	—	—	—	—	—	—	2 48
6	— 8 9	70 1	43	67 8	64 8	32 9	33 6	28 3	8 9	2 8	— 2 1	—	—	—	2 84
7-A	— 8 9	70 1	43	—	—	—	—	—	—	—	—	—	—	—	1 16
8-A	— 8 9	70 1	43	—	—	—	—	—	—	—	—	—	—	—	0 89
C-2	— 8 9	70 1	43	—	—	—	—	—	—	—	—	—	—	—	3 0
9	— 8 9	70 8	42	65 9	62 9	28 1	28 0	23 0	4 4	— 1 4	— 6 1	—	—	—	0 43
10	— 8 9	70 8	42	—	—	—	—	—	—	—	—	—	—	—	—
11	— 8 9	70 8	42	66 9	62 3	24 2	27 9	21 1	5 1	— 0 5	— 6 6	—	—	—	0 16
12	— 8 9	70 8	42	—	—	—	—	—	—	—	—	—	—	—	—
C-3	— 8 9	70 8	42	72 3	68 2	—	29 4	—	1 1	— 3 5	—	—	—	—	0 17
13	— 8 9	70 4	42	68 3	64 5	25 5	23 2	20 0	5 9	0 7	— 5 9	—	—	—	0 23
14	— 8 9	70 4	42	—	—	—	—	—	—	—	—	—	—	—	—
15	— 8 9	70 4	42	—	—	—	—	—	—	—	—	—	—	—	—
16	— 8 9	70 4	42	—	—	—	—	—	—	—	—	—	—	—	0 04
C-4	— 8 9	70 4	42	—	—	—	—	—	—	—	—	—	—	—	0 19
Test Period No. 2—114 Hours															
1-A	—11 4	69 7	41	—	—	—	—	—	—	—	—	0 0	—	1 64	1 25
2	—11 4	69 7	41	64 2	58 8	55 0	47 8	36 4	30 4	12 9	1 7	—	—	0 53	0 93
3-A	—11 4	69 7	41	—	—	—	—	—	—	—	—	—0 23	0 94	1 47	1 51
4-A	—11 4	69 7	41	—	—	—	—	—	—	—	—	—0 10	2 04	1 37	1 12
C-1	—11 4	69 7	41	—	—	—	—	—	—	—	—	—	0 62	—	—
5-A	—11 4	70 4	40	—	—	—	—	—	—	—	—	0 06	2 98	2 69	3 03
6	—11 4	70 4	40	68 1	64 9	31 9	33 5	28 5	7 9	0 9	— 4 4	0 04	—	2 74	2 66
7-A	—11 4	70 4	40	—	—	—	—	—	—	—	—	—0 06	—	1 02	0 86
8-A	—11 4	70 4	40	—	—	—	—	—	—	—	—	0 14	0 78	0 65	0 37
C-2	—11 4	70 4	40	—	—	—	—	—	—	—	—	—	0 16	3 16	3 35
9	—11 4	70 0	40	65 4	62 4	26 6	26 6	22 6	2 1	— 3 8	— 8 2	0 05	0 32	0 46	0 51
10	—11 4	70 0	40	—	—	—	—	—	—	—	—	—	0 48	—	—
11	—11 4	70 0	40	66 4	62 0	24 6	29 5	20 7	1 7	— 3 8	— 9 2	—0 05	0 16	0 23	0 33
12	—11 4	70 0	40	—	—	—	—	—	—	—	—	—	0 32	—	—
C-3	—11 4	70 0	40	71 9	67 4	—	27 8	—	— 1 1	— 5 8	—	—	0 00	0 17	0 18
13	—11 4	70 4	40	68 3	64 7	25 1	25 6	20 5	3 6	— 2 5	— 8 6	—0 18	0 32	0 17	0 09
14	—11 4	70 4	40	—	—	—	—	—	—	—	—	—	0 16	—	—
15	—11 4	70 4	40	—	—	—	—	—	—	—	—	—	0 32	—	—
16	—11 4	70 4	40	—	—	—	—	—	—	—	—	—0 44	0 16	0 02	0 00
C-4	—11 4	70 4	40	—	—	—	—	—	—	—	—	—	0 16	0 14	0 09

Table 5—Test No. 3—Data and Results

Wall No.	Photograph Taken Fig. No.	Test Period No. 1—80 Hours					Test Period No. 2—120 Hours				
		Cold Room F	Test House		Inside Surface Temperature of Sheathing F	Moisture Gain on Aluminum Panel Grams per Sq Ft per 24 Hours	Cold Room F	Test House		Inside Surface Temperature of Sheathing F	Moisture Gain on Aluminum Panel Grams per Sq Ft per 24 Hours
			Dry-Bulb F	Per Cent R. H.				Dry-Bulb F	Per Cent R. H.		
1-A		— 7 4	70 3	40	9 8	1 11	— 7 7	70 0	40	9 7	0 31
2[a]	17	— 7 4	70 3	40	29 5	0 13	— 7 7	70 0	40	28 5	0 46
3-A		— 7 4	70 3	40	10 7	0 98	— 7 7	70 0	40	10 6	0 23
4-A		— 7 4	70 3	40	10 5	1 31	— 7 7	70 0	40	10 7	0 33
C-1A		— 7 4	70 3	40	9 6	2 56	— 7 7	70 0	40	9 2	2 45
5-A		— 7 4	70 0	40	3 8	0 0	— 7 7	70 4	40	4 3	0 0
6[a]	18	— 7 4	70 0	40	10 3	2 26	— 7 7	70 4	40	10 2	2 34
7-A[a]	24	— 7 4	70 0	40	11 5	0 75	— 7 7	70 4	40	11 8	0 22
8-A		— 7 4	70 0	40	13 5	0 82	— 7 7	70 4	40	13 4	0 22
C-2[a]	28	— 7 4	70 0	40	5 9	2 56	— 7 7	70 4	40	6 9	2 68
9		— 7 4	70 5	39 5	7 3	0 24	— 7 7	69 9	40	7 6	0 21
10[a]	20	— 7 4	70 5	39 5	9 1	0 20	— 7 7	69 9	40	9 2	0 19
11		— 7 4	70 5	39 5	6 2	0 00	— 7 7	69 9	40	5 8	0 0
12[a]	21	— 7 4	70 5	39 5	8 7	0 09	— 7 7	69 9	40	9 1	0 06
C-3		— 7 4	70 5	39 5	3 8	0 03	— 7 7	69 9	40	4 2	0 02
13		— 7 4	70 1	40	6 8	0 13	— 7 7	70 4	40	7 5	0 11
14[a]	22	— 7 4	70 1	40	9 1	0 13	— 7 7	70 4	40	9 5	0 12
15		— 7 4	70 1	40	8 0	0 0	— 7 7	70 4	40	8 4	0 0
16[a]	23	— 7 4	70 1	40	8 4	0 0	— 7 7	70 4	40	8 8	0 0
C-4[a]	29	— 7 4	70 1	40	5 8	0 13	— 7 7	70 4	40	6 8	0 15
17		— 7 4	70 0	40	7 2	0 26	— 7 7	70 0	40	8 1	0 46
18[a]	19	— 7 4	70 0	40	8 9	0 42	— 7 7	70 0	40	9 3	0 56
19		— 7 4	70 0	40	8 9	2 36	— 7 7	70 0	40	9 6	2 71
20		— 7 4	70 0	40	9 7	2 48	— 7 7	70 0	40	10 2	3 02
C-5		— 7 4	70 0	40	7 1	4 12	— 7 7	70 0	40	7 4	2 96
21		— 7 4	70 2	40	11 3	1 77	— 7 7	69 9	40	10 7	3 08
24		— 7 4	70 2	40	11 0	0 88	— 7 7	69 9	40	11 1	0 15
C-6		— 7 4	70 2	40	8 7	1 31	— 7 7	69 9	40	9 6	2 45

[a]Photographs taken at the end of test.

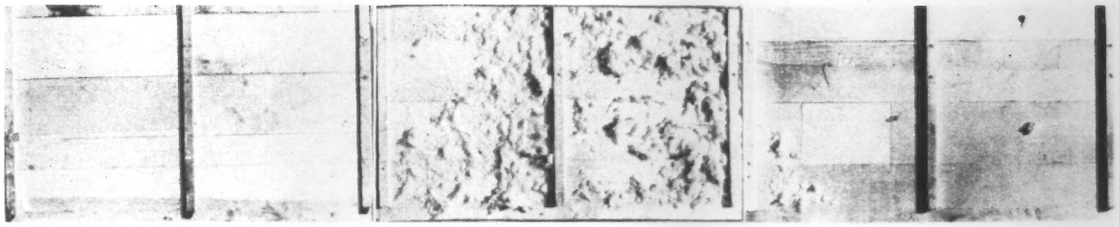


Fig. 12—Wall No. 4 Fig. 13—Wall No. 5 Fig. 14—Wall No. 10

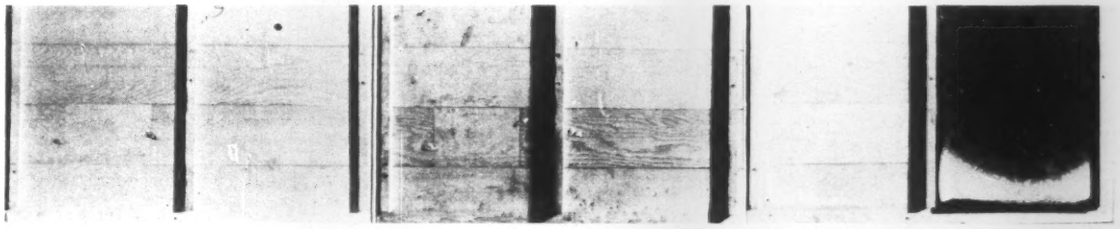


Fig. 15—Wall No. 12 Fig. 16—Wall No. 16 Fig. 17—Wall No. 2

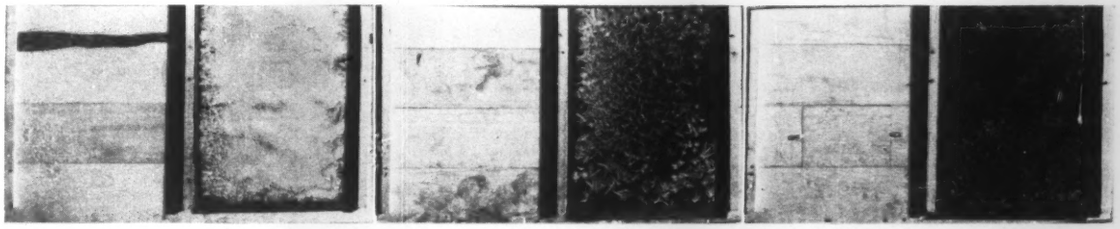


Fig. 18—Wall No. 6 Fig. 19—Wall No. 18 Fig. 20—Wall No. 10

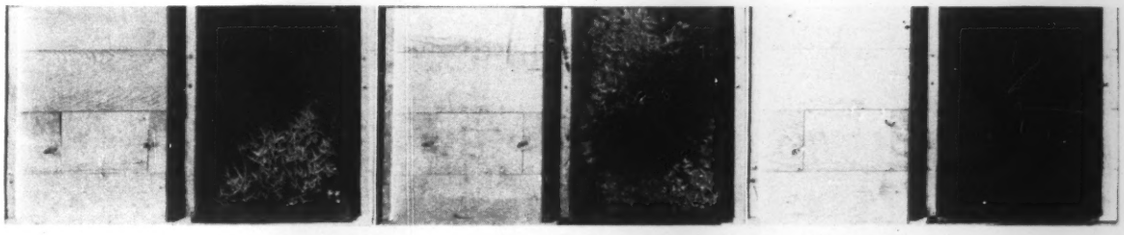


Fig. 21—Wall No. 12 Fig. 22—Wall No. 14 Fig. 23—Wall No. 16

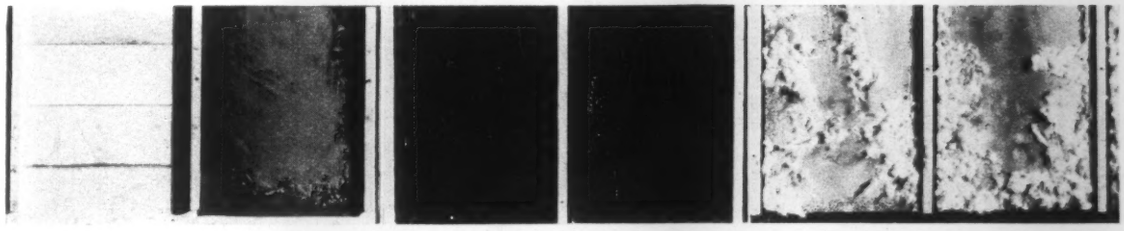


Fig. 24—Wall 7A Fig. 25—Ceiling No. C-1 Fig. 26—Ceiling No. C-2

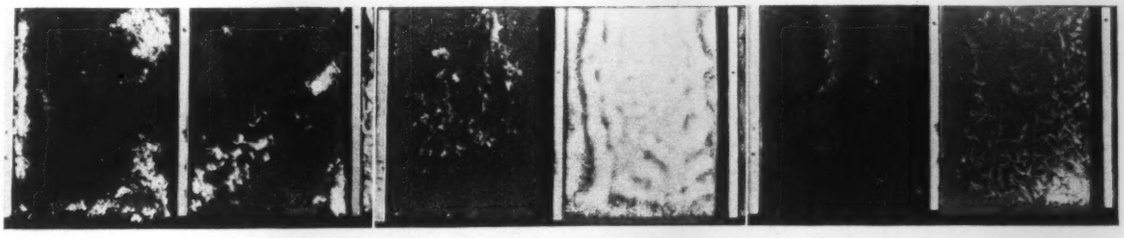


Fig. 27—Ceiling No. C-4 Fig. 28—Ceiling No. C-2 Fig. 29—Ceiling No. C-4

ture collected on the aluminum panel and that collected on the sheathing for similar sections of a wall of the same construction and under the same test conditions. The greatest error in the collection of moisture by this method appeared to be in the uninsulated walls. In these cases the inner surfaces of the sheathing were not far from freezing temperatures and the moisture was not retained on the surface of the sheathing as frost but seemed to penetrate and pass through. The difference between the vapor pressure of the air on the two sides of sheathing was greater for the uninsulated wall than it was for the insulated walls. Thus the vapor which did travel through the inner finish had a greater tendency to travel through the outer finish and not be collected. In these tests the temperature of the inner surface of the outside sheathing has been taken as a measure of the dew-point temperature of the air within the space between the studs. This is sufficiently accurate for comparative purposes, but from an investigation which is now under way it appears that the actual dew-point temperature of the air in the space between the studs is a considerable amount higher than the surface temperatures of the sheathing and that for absolute results the vapor pressure must be measured. While there is a great deal more test work necessary to determine absolute values for various types of construction, and to find the most practical applications for different outside weather and inside air conditions, the work has been carried far enough to show that there are practical solutions of the condensation problem and that when conditions are thoroughly understood it will be only a matter of selecting material and types of construction to insure satisfactory results. The following comparisons taken from the test data are of interest as indicating what may be expected under various combinations of construction and operating conditions.

Different Types of Construction Compared

Table 1 gives the details of the walls as tested, and Table 2 gives the details of the ceilings. The test data for tests Nos. 1, 2 and 3 are shown in Tables 3, 4 and 5, respectively. As a matter of convenience, Table 6 has been arranged showing the results obtained for the different walls during the second period of the third test. These were all tested under the same conditions and the results showing condensate in grams per square foot per 24 hours were taken as the amount collected on the aluminum sheet placed next to the sheathing in the center stud section. Thus the values are all on the same basis and show a definite comparison between various types of construction. The results for the uninsulated wall No. 2 compared with the insulated walls Nos. 6 and 19 show clearly what may happen when insulation is added to a wall without giving any consideration to the relative vapor resistance of the different sections of that wall. These walls were of a common type of construction with metal lath and plaster on inside surface, and sheathing, building paper, and lap siding on the outside surface. This construction gives a low vapor resistance for the inside surface and a relatively high resistance at the outer surface. Thus the vapor density may build up at the outer surface. In addition to this the inside surface finish of the wall has but very little insulating value and for the uninsulated wall the heat travels through to the outside finish and raises the inside surface of the sheathing to a much higher temperature than will be the case when insulation is used. Thus the possibility of condensation of moisture on the sheathing of the uninsulated wall is reduced, but at the same time the heat losses through the wall are quite high.

The photographs of Figs. 17 and 18 show the conditions which existed after this test. The uninsulated wall, Fig. 17, shows both the wood sheathing and aluminum panel at center section to be free from moisture excepting at the lower part of the wall through which this photograph was taken. The accumulation of frost extended approximately 6 in. from the bottom on the aluminum panel and around the lower outside corner of the wood sheathing. For the insulated wall, Fig. 18, the frost was rather uniformly distributed over the entire aluminum panel and also over the wood sheathing in the exposed section. That part of the wood sheathing shown in the photograph has slightly heavier frost formation than the average for the entire area. The weight of frost collected per square foot of aluminum panel for the uninsulated wall No. 2 was 0.46 grams per 24 hours, and for the insulated wall No. 6 it was 2.34 grams per 24 hours. The results for these two walls show a condition often met in practice and misinterpreted.

The advantage to be gained by adding vapor barriers

Table 6—Relative Condensation for Different Types of Wall Construction Under Similar Test Conditions

Type of Wall and Insulation	Special Vapor Barrier	Wall No.	Photo After Test Fig. No.	Test No. and Period No.	Cold Room Temp. F	Surface Temp. of Sheathing F	Condensate Grams Per Sq Ft Per 24 Hours
Standard, No insulation	None	2	17	3-2	—7 7	28 5	0 46
Standard with 3⅝ in. rock wool	None	6	18	3-2	—7 7	10 2	2 34
Standard with 3⅝ in. rock wool	None	19	—	3-2	—7 7	9 6	2 71
Standard with 3⅝ in. rock wool	50 lb. Bldg. Paper,[a] lapped joints	17	—	3-2	—7 7	8 1	0 46
Standard with 3⅝ in. rock wool	A[a] lapped joints	18	19	3-2	—7 7	9 3	0 56
Standard with 3⅝ in. rock wool	B[a] lapped joints	9	—	3-2	—7 7	7 6	0 21
Standard with 3⅝ in. rock wool	A[a] sealed joints	10	20	3-2	—7 7	9 2	0 19
Standard with 3⅝ in. rock wool	B[a] sealed joints	11	—	3-2	—7 7	5 8	0 0
Standard with 3⅝ in. rock wool	2 coats flat white paint	12	21	3-2	—7 7	9 1	0 06
Standard with 3⅝ in. rock wool	2 coats aluminum paint	13	—	3-2	—7 7	7 5	0 11
Standard with 3⅝ in. rock wool	1 coat size, 2 coats flat white paint	14	22	3-2	—7 7	9 5	0 12
Standard with 3⅝ in. rock wool	2 coats size	15	—	3-2	—7 7	8 4	0 0
Standard with 3⅝ in. rock wool	1 coat size, 2 coats aluminum paint	16	23	3-2	—7 7	8 4	0 0
Standard with 3⅝ in. rock wool	1 coat asphalt paint	1-A	—	3-2	—7 7	9 7	0 31
Standard with 3⅝ in. rock wool	2 in. hole in sheathing only	3-A	—	3-2	—7 7	10 6	0 23
Standard with 3⅝ in. rock wool	None	4-A	—	3-2	—7 7	10 7	0 33
Standard with 3⅝ in. rock wool	No Bldg. Paper between sheathing and siding	7-A	24	3-2	—7 7	11 8	0 22
Standard with 3⅝ in. rock wool	4-1 in. vent holes in each stud section	8-A	—	3-2	—7 7	13 4	0 22
Standard with 3⅝ in. rock wool		24	—	3-2	—7 7	11 1	0 15
Standard with 3⅝ in. rock wool		21	—	3-2	—7 7	10 7	3 08
No Bldg. Paper between sheathing and siding 3⅝ in. rock wool		20	—	3-2	—7 7	9 6	3 06
Standard with 3⅝ in. rock wool		5-A	—	3-2	—7 7	4 3	0 0

[a] All vapor barriers placed between metal lath and studs.
A = 30, 30, 30 duplex paper, 2 sheets 30 lb paper cemented together with 30 lb asphalt.
B = Asphalt impregnated and surface coated sheathing paper, glossy surfaced and weighing 50 lb per roll of 500 sq ft.

to the inside surface of a wall is shown by the various photographs and weights of condensate indicated in Table 6 for different types of barriers. A 50 lb standard building paper placed between the metal lath and studs of insulated walls Nos. 17 and 18 showed 0.46 and 0.56 grams of condensate respectively as compared with 2.34 and 2.71 for insulated walls Nos. 6 and 19 respectively, which were of the same construction but with no vapor barrier between the metal lath and studs. The photograph of Fig. 19 shows the condition of the aluminum panel for wall No. 18 after the test. The sheathing showed no signs of condensation, and that on the aluminum panel was very light as indicated by the weight.

Walls Nos. 9 and 10 and photograph of Fig. 20 show the effect of placing the barrier *A*, known as 30,30,30 duplex paper, between the metal lath and studs. In this case there were 0.21 and 0.19 grams of condensate for 24 hours. No frost appeared on the wood sheathing, and that on the aluminum panel was very light and scattered. The effect of placing an asphalt impregnated and glossy surfaced coated sheathing paper, with joints lapped but not sealed, between the metal lath and studs of an insulated wall is shown by the results for walls Nos. 11 and 12, and photograph of Fig. 21. For wall No. 11 the condensate was zero and for Wall No. 12, 0.06 grams per 24 hours. The photograph of wall No. 12 shows a very slight amount of condensate collected at the lower part of the aluminum panel but none on the sheathing. A slightly better result was obtained when the two vapor barriers, *A* and *B*, were sealed at the joints, as shown in walls Nos. 13 and 14 and photograph of Fig. 22, for barrier *A*, and walls Nos. 15 and 16, photograph of Fig. 23, for vapor barrier *B*. As shown by the weight and in Fig. 22, when barrier *A* was used a slight amount of frost was collected on the aluminum panel, mostly at the bottom section. The inner surface of the sheathing did not show any signs of frost. When barrier *B* was used no frost was collected on either the aluminum panel or the inside surface of the sheathing. Thus this was an effective barrier to the passage of moisture.

The effect of adding a moisture barrier to the inside surface finish of a wall is shown in the results for walls Nos. 1-A, 3-A, 4-A, 7-A, 8-A and 24 of tests 3, second period. The amount of condensation ranged from 0.15 grams per 24 hours for one coat of asphalt paint to 0.33 for one coat size and two coats white flat paint. Two coats size, 2 coats aluminum paint and one coat of size with 2 coats of aluminum paint all showed 0.22 grams of condensate. It appears that a very effective barrier to the passage of vapor may be applied as an inside surface finish to the wall. However, more experimental work is needed to determine the best barriers to select.

The effect due to ventilating the sheathing to the outside has received only a small amount of attention thus far. In Wall No. 5-A, test 3, two 1-in. vent holes were drilled through the sheathing and siding at the top and bottom of the center stud space. This resulted in zero condensation but it also lowered the surface temperature of the sheathing. In some of the walls on other tests ¼ in. holes were drilled and were found too small to be effective. It is evident that such ventilation may be established, but more investigation is required to find the practical limits.

Results for Ceiling Construction

The ceilings were of somewhat different construction than the side walls in that a 50 lb building paper was placed between the top of the joists and the outside sheathing. Three coats of white lead paint were applied directly to the sheathing with no other exterior finish. Some of the ceilings were constructed without insulation and others were constructed with different types of fill insulation. The construction of the ceilings is shown in Table 2; the comparative test results for those used in test 3 period 2 are shown in Table 7, and the conditions after certain tests are shown in the photographs, Figs. 25 to 29, inclusive.

The building paper on the cold side of the insulation acted as a moisture barrier and collected much more frost than was collected on the sheathing of similarly constructed side walls. It appeared to be a much better barrier to the passage of moisture at low temperatures than it was at high temperatures. Thus in the uninsulated ceiling it was evident that some of the vapor which passed into the space between the joists was either escaping through the building paper which was at a fairly warm temperature or else it was leaking through the joints around the edges of the ceiling. In wall No. 18, Fig. 19, in which a 50 lb building paper was used as a vapor barrier between the metal lath and studs, a considerable amount of vapor passed through the paper as shown by the weight of frost condensed on the inside of the sheathing. In ceiling C-2, Fig. 28, in which insulation was used, an aluminum sheet was placed between central joists with the building paper exposed in the two outer joist sections. After test 3, period 2, the building paper on the two outside joist sections was removed and weighed and then dried and re-weighed to determine the amount of frost collected. The amount was found to be exactly the same as that collected by the central aluminum panel, indicating that the building paper was a rather effective barrier at the low temperatures. This, however, cannot be taken as conclusive proof as some of the test data showed higher vapor travel through the outside stud section than through the center stud section, possibly due to lower temperatures in the outside exposed corner section of the wall.

Fig. 25 shows the typical condition of the paper on the under side of the sheathing of an uninsulated wall after any of the test periods. In the longer periods there was slightly more ice and frost formation around the edges than there was in the shorter test. The central sections were always clear, and the formation of ice and frost around the edges was apparently due to the cooling effect from the exposed corners. In all cases of either ceilings or side walls where frost was formed there were heavier formations along the surfaces next to the exposed corners, and in some cases it was heavier along the central section between the studs of the insulated walls with clear lines next to the studs.. This formation was due to the fact that the heat transfer through the studs was greater than through the insulated sections between the studs, therefore giving a slightly warmer section of the sheathing next to the studs with less chance for the formation of frost.

A photograph of ceiling No. C-2 shown in Fig. 26 after 460 hours test period is typical of the results ob-

Table 7—Relative Condensation for Different Types of Ceiling Construction under Similar Test Conditions

Type of Ceiling and Insulation	Special Vapor Barrier	Ceiling No.	Photo After Test Fig. No.	Test No. and Period No.	Cold Room F	Surface Temp. of Sheathing	Condensate Grams Per Sq Ft Per 24 Hours
Standard with 3⅝ in. rock wool	None	C-2	28	3-2	—7.7	6.9	2.68
Standard with 3⅝ in. expanded aluminum magnesium silicate	None	C-1-A	—	3-2	—7.7	9.2	2.45
Standard with 3⅝ in. redwood bark fiber	None	C-5	—	3-2	—7.7	7.4	2.96
Standard with 3⅝ in. sawdust	None	C-6	—	3-2	—7.7	9.6	2.45
Standard with 3⅝ in. rock wool	B[a] lapped joints	C-3	—	3-2	—7.7	4.2	0.02
Standard with 3⅝ in. rock wool	B[a] sealed joints	C-4	29	3-2	—7.7	6.8	0.15

[a]All vapor barriers placed between metal lath and studs.
B = Asphalt impregnated and surface coated sheathing paper, glossy surfaced and weighing 50 lb per roll of 500 sq ft.

tained for all insulated ceilings and walls with no moisture barriers on the warm side. The heavier frost formation in comparison with the uninsulated walls is accounted for by the fact that the exterior surface is much colder for these walls than for the uninsulated walls, and therefore condenses more of the vapor out of the space between the studs, which in turn lowers the vapor pressure in this space and gives a greater vapor pressure drop through the inside surface, with a corresponding increase in the vapor travel through this section of the wall.

The values in Table 7 observed for test 3, period 2 show the relative amounts of vapor traveling through the plaster of ceilings with and without vapor seal construction for different types of fill insulation. In the first four ceilings four different types of insulation were used and no vapor seal applied. The amounts of condensation are substantially the same. In the last two, asphalt impregnated and surface coated sheathing paper with glossy surfaces was applied on the studs underneath the metal lath and for these ceilings the amount of condensate was practically negligible.

Moisture Change in Insulation

In several of the tests the rock wool was weighed when installed and taken out and weighed immediately after the test to determine whether or not there was any absorption of moisture. The results for these weights are shown for several walls in Tables 3 and 4. While there might have been some small losses in the material between weighing periods, every precaution was taken to prevent this and in the majority of cases a decrease in weight is shown, and in no case is there an increase greater than what might be expected by experimental error. These tests show conclusively that the rock wool did not absorb moisture from the vapor in the air during the tests.

Conclusions

There is no doubt but that the condensation of moisture in walls is a troublesome problem with many types of construction under adverse conditions, and that in the past many difficult situations have arisen due to a lack of understanding of the true nature of the problem and to the fact that different building materials have been combined without regard to the effect of these combinations on the thermal and vapor properties of a wall. Since the temperatures of the various parts of a wall are controlled by the type and amount of insulation used and the vapor densities in the corresponding parts are controlled by the vapor barriers used, the transmission of heat and vapor through a wall must be considered together, and in most cases an intelligent combination of insulation and vapor barriers will eliminate any chance of trouble. A fundamental principle often overlooked is that a vapor barrier should be placed on the warm side and not on the cold side of a wall.

While there is still much research work to be done concerning practical vapor barriers and the limiting conditions under which they may be used, the results obtained thus far prove the practicability of solving the problem by applying high vapor resistance materials to the inner section and low vapor resistance materials to the outer section of the wall. The problem is not to be feared after it is once understood. It is the intention to continue the investigation not only to study the individual vapor barriers and their limiting conditions but also to consider all of the problems involved in practical house construction by the use of large scale test houses within the cold room.

Acknowledgments

This research program has been initiated by the *National Rock and Slag Wool Association* in cooperation with the University of Minnesota to make a thorough study of the elements found in the condensation problem and to determine the most practical solution from the builders' point of view.

The authors wish to acknowledge the cooperation and assistance of the Technical Committee of the *National Rock and Slag Wool Association* in the entire program. This committee, consisting of Messrs. Jan S. Irvine, Chairman; W. W. Collins; E. W. McMullen; C. L. Neumeister; Earl Williams, and the late Dr. B. Townshend have given freely of their time and efforts to make this work possible.

The authors and the Committee acknowledge the valuable assistance and cooperation given by L. V. Teesdale, Senior Engineer, and his associates, of the Forest Products Laboratory, U. S. Department of Agriculture, Madison, Wisconsin. Mr. Teesdale has met several times with the Committee and has contributed liberally from the experience accumulated in an extended study of this problem at the Forest Products Laboratory.

M. L. Fergestad Joins Bark Products Division

M. L. Fergestad has become associated with the Pacific Lumber Company, Minneapolis, Minn., in the Bark Products Division. As field engineer he will be engaged in sales and technical work in connection with the company's insulation material.

Ventilating the Lincoln Vehicular Tunnel

By Charles W. Murdock* (*NON-MEMBER*), New York, N. Y.

WITH the opening of the South Tube of the Lincoln Tunnel in December, 1937, a second great motor vehicle tunnel under the Hudson River was made available for the ever increasing traffic between New Jersey and New York. Located in Manhattan's mid-town area the new tunnel is the third trans-Hudson River vehicular crossing operated by the Port Authority, the others being the Holland Tunnel, completed in 1927, and the George Washington Bridge, opened in 1932.

Ground was broken on May 17, 1934, for this crossing, which was known as the Mid-town Hudson Tunnel until officially named the Lincoln Tunnel in April, 1937.

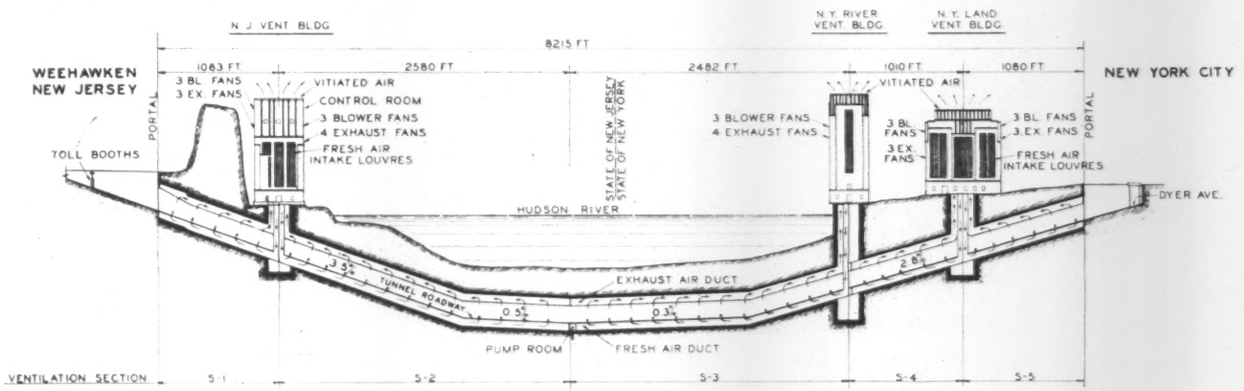


Fig. 1—Profile view of south tube

It will have two separate tubes when completed, each tube providing for two-lane traffic in one direction. Until the North Tube is completed, the South Tube is to carry two-way traffic and thereafter will be used for traffic East-bound from New Jersey while the North Tube will carry the West-bound traffic to New Jersey.

The strategic location of the Lincoln Tunnel should provide considerable relief for the serious traffic problems of New York City, since with the completion of the Queens Tunnel, now being built under the East River at 38th Street by the City of New York, and the proposed crosstown vehicular tunnel through the heart of the Mid-town section, a direct underground highway from New Jersey and the West and South to Long Island will be provided, with surface connections on both east and west sides of the City.

While in general design and construction details the new tunnel is similar to the Holland Tunnel, its larger tube diameter provides a wider roadway and air ducts of larger cross-section. The principal divergence in the general scheme of the ventilating systems of the two tunnels is that the Lincoln Tunnel has three ventilation buildings, two in New York and one in New Jersey, instead of four as were built for the Holland Tunnel, which has two ventilation buildings on each side of the river.

General Plan of Tunnel

The Lincoln Tunnel passes under the Hudson River as shown in Fig. 1, from West 39th Street in Manhattan to the New Jersey shore where it bores through Kings Bluff and then turns southward in a large radius curve emerging on a plaza located between the Hudson County Boulevard East and Park Avenue in Weehawken. Tolls will be collected at toll booths located on this plaza. Connections with Union City, North Bergen and neighboring towns are provided. A new express highway which loops to the north and west from the plaza is being constructed and when completed will connect with Routes 1 and 3 of the New Jersey Highway system.

The New York Plaza of the South Tube is located in the westerly half of the block bounded by Ninth and Tenth Avenues and West Thirty-eighth to West Thirty-ninth Streets. Access to the plaza is provided by Dyer Avenue, a new street between Ninth and Tenth Avenues extending from Thirty-fourth Street to Forty-second Street.

The under river tubes have an outside diameter of 31 ft, as indicated in Fig. 2, which is 1 ft 6 in. larger than the tubes of the Holland Tunnel and provides a roadway width between curbs of 21 ft 6 in. which is 1 ft 6 in. wider than the roadways in the older tunnel. Roadway headroom throughout is approximately 13 ft 7 in. which permits a safe operating clearance of 13 ft.

The South Tube is 8215 ft long between portals and consists of 240 ft of steel bent construction and 785 ft of steel lined circular rock tunnel on the New Jersey side, 5920 ft of cast iron lined shield driven tunnel under the river and 450 ft of steel lined rock tunnel and 560 ft of steel bent tunnel to the New York Portal. The remaining 260 ft are contained within the limits of the ventilating shafts. The North Tunnel, now under con-

*Mechanical Engineer, The Port of New York Authority.

For presentation at the 44th Annual Meeting of the American Society of Heating and Ventilating Engineers, New York, N. Y., January, 1938.

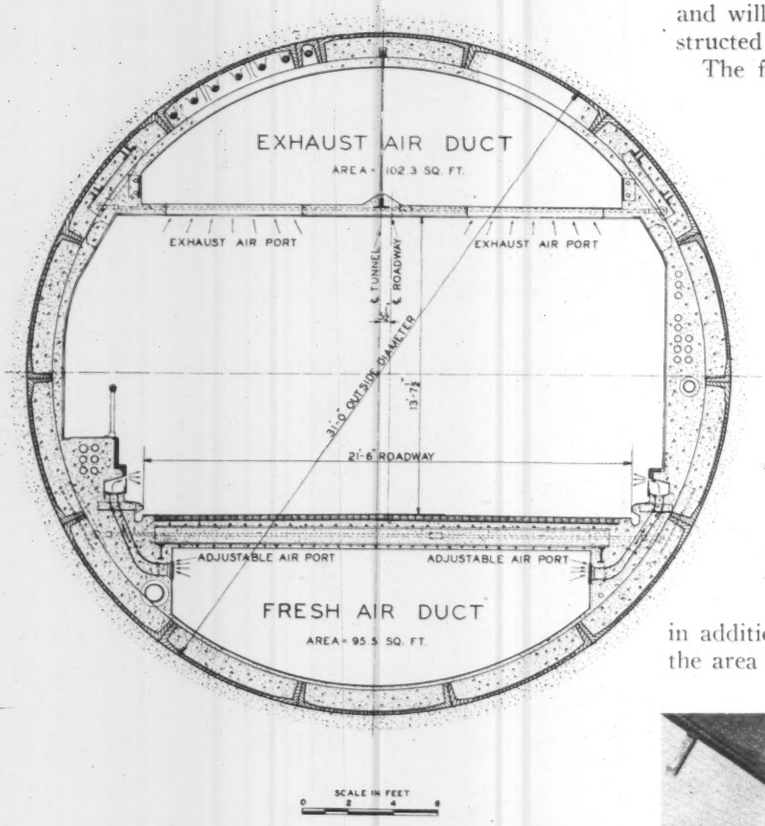


Fig. 2—Cross-section of river tube

struction, will have a total length between portals of about 7400 ft as its New York Portal will be located about one block nearer the river.

The maximum depth of the top of the tubes below mean high water is 75 ft and the roadway is 97 ft below the river at that point. The two tubes in the river section are to be 75 ft apart, center to center, and will have a minimum covering of 20 ft of silt. Among the more interesting constructional features of the tunnel might be mentioned the special lighting arrangements, the glass tile ceiling and the vitrified brick roadway surfacing, as shown in Fig. 3, the latter two items being new departures in vehicular tunnel design. The following tabulated data are given as being of general interest:

Total length (South Tube)	8,215 ft
Length of River Section Center to Center of Ventilation Buildings	5,062 ft
Number of Ventilation Buildings	3
Number of Fresh Air Fans—South Tube	15
Number of Exhaust Air Fans—South Tube	17
Maximum Fresh Air Supplied—South Tube	1,736,000 cfm
Maximum Operating Horsepower of Fan Motors—South Tube	2,237

Description of the Ventilating System

The ventilating system of the South Tube is essentially the same as that so successfully employed in the Holland Tunnel, and is generally known as the *Transverse Flow* type. This system has been used in The Detroit-Windsor Tunnel, the George A. Posey Tunnel in Oakland, Calif., the Boston Traffic Tubes, and the Antwerp Tunnel, and will be used in the Queens Tunnel now being constructed under the East River.

The fresh air from the blower fans flows through a duct below the roadway slab, which is illustrated in Fig. 4. Air is introduced into the roadway area through a continuous expansion chamber and a narrow adjustable slot located above each curb and connected to the duct below by flues 3½ ft wide and 6 in. high, spaced from 10 to 15 ft apart throughout the tunnel length. A uniform distribution of the fresh air throughout the tunnel length is obtained by adjusting the asbestos slide in each flue inlet according to the static pressure in the duct.

The exhaust duct is formed by the ceiling slab and the upper part of the tube shell, the vitiated air being withdrawn through adjustable cast iron ports, placed in pairs, one on each side of the centerline, and spaced at intervals of from 10 to 20 ft longitudinally. These exhaust port castings vary in size, and in addition have adjustable stainless steel slides so that the area of each port opening can be set as required by

Fig. 3—Interior view of tunnel roadway

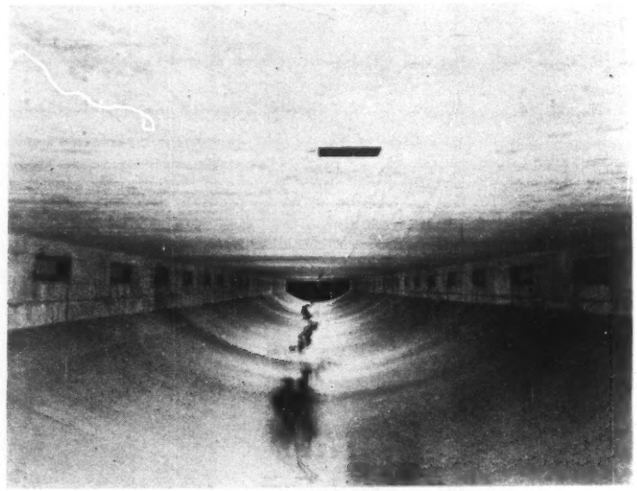


Fig. 4—Sectional view of fresh air duct

the static pressure in the exhaust duct at the point where the port is located.

The tunnel is divided into five ventilation sections, each section being served by one fresh air duct and one exhaust air duct which ascend vertically at the shafts to the blower and exhaust fans located in the ventilation buildings above. The exhaust air ducts lead to air tight rooms in which the exhaust fans are placed, and the vitiated air is discharged to the outside atmosphere above the roof of the building.

The chief advantage of this method of ventilation is the practically complete elimination of longitudinal air currents. It is felt that a longitudinal flow of air in the space used by vehicles presents an undue hazard if the tunnel is one used by a large volume of traffic and the velocity of flow is such as would rapidly spread smoke or other fumes. The chief disadvantages are the higher initial and operating costs.

Experience in the operation and maintenance of the Holland Tunnel has resulted in improvements in the ventilating equipment and its arrangement in the buildings, and has shown how careful attention to the many minor details simplifies the maintenance problem. In the design of the various parts of the duct systems particular attention has been directed to the improvement in details of construction and the reduction of resistance losses. All elbows have liberal radii and where conditions warrant it they are provided with splitting vanes. Changes in area or shape are as gradual as possible, especially in expanding ducts. The areas of the exhaust ducts at the point where they enter the exhaust fan rooms are large in order that the expansion loss at those points will be small. Access to all tunnel ducts has been provided at the ventilating shafts making it unnecessary to use the manholes in the roadway. The adjustable slide of the fresh air ports is accessible from the duct so that it is unnecessary to remove the expansion chamber face plate to make a change in the port area.

Air Requirements for Ventilation

The determination of the amount of air required to ventilate the tunnel was based on the extensive Tunnel Gas Investigations conducted by the U. S. Bureau of Mines in 1920 and on similar tests made in 1934 at the Holland Tunnel which indicated that there has been comparatively little change in the quantity of carbon monoxide produced by motor vehicles since the earlier tests were conducted. The Bureau of Mines tests have been fully described in a bulletin[1] and summaries have been given in other literature. On the basis of these tests and actual experience in tunnel operation, the maximum rates of fresh air supply for the new tunnel were determined as follows:

Two-way Traffic

Section on Grades.....200 cfm per linear foot of tunnel
Level Sections .....225 cfm per linear foot of tunnel

One-way Traffic

Sections on Up-grades.....275 cfm per linear foot of tunnel
Section on Down-grades.....150 cfm per linear foot of tunnel
Level Sections .....225 cfm per linear foot of tunnel

[1]Ventilation of Vehicular Tunnels, by A. C. Fieldner, Yandell Henderson, J. W. Paul, R. R. Sayers, et al. (A. S. H. V. E. JOURNAL, January-December, 1926).

These rates provide a margin of safety above the actual requirements for the safe ventilation of the tunnel at its maximum traffic capacity. However, in case of a fire or traffic jam this excess air supply will insure the rapid removal of the smoke and gases.

Since the South Tube will be used for two-way traffic until completion of the North Tube, when it will be used only for one-way traffic, all calculations used in the design had to be based upon the two different rates of air supply. When the South Tube is converted to one-way traffic operation, new settings of the fresh and exhaust air ports will be required. In addition the bulkhead between the two exhaust ducts under the river will be moved 300 ft to the east to properly equalize the performance of the fans connected to these ducts.

In all computations of air flow the density of the fresh air has been taken as 0.08 lb per cubic foot and for the exhaust air 0.0784 lb per cubic foot, these factors having been determined from a study of the U. S. Weather Bureau Records for a period covering eight years. The density of the exhaust air has been calculated by assuming a 10 deg rise in temperature of the air in the tunnel due to the heat in the motor exhaust gases.

Air Measurement Apparatus

To provide for the testing of the fans in actual operation and to check the velocity and static pressures in the ducts against the calculated values used in the design, permanent piping connections for the air measurement equipment have been incorporated in the walls of the fresh air and exhaust ducts. These measurement stations which are spaced at distances varying from 300 to 400 ft throughout the tunnel length, consist of piezometer rings with plates set in the sides of the ducts and a second pipe terminating at the duct in such a way that a pitot tube can be located at the center of gravity of the duct area. The piping extends to a box in the wall above the sidewalk where connections to portable manometers can be made by means of rubber tubing. In addition the fresh air duct has a series of ferrules set in the roadway slab while the exhaust air duct has ferrules in the tunnel ceiling. Pitot tubes can be inserted through these ferrules and readings taken at the proper points of the duct cross-section as is done in testing fans. From these readings the volume of air flowing in the ducts and the static pressures can be calculated and also a center constant established by means of which future determinations of the volume of air flowing in the ducts can be made without any interruption of traffic. This arrangement, in conjunction with the carbon monoxide recorders, makes it possible at any time and under any traffic condition to make a field check on the amount of carbon monoxide being liberated in the tunnel.

The Ventilation Buildings

The location of the three ventilation buildings housing the fan equipment and providing the necessary air intakes and outlets for the system was determined by a study of the various factors involved. Comparative estimates of initial and operating costs indicated that two buildings for the entire system would be more economical than either three or four, provided that they could be placed at fairly definite positions in relation to the

tunnel length. The required position in New York was entirely impractical and in New Jersey the probability of construction difficulties made it seem advisable to locate that building further west where a rock foundation was available. These considerations made the three building layout the obvious solution.

The New York Land Ventilation Building is located at 39th Street and Eleventh Avenue and is built on foundations which extend below the tube and form the shaft to the tunnel. This building is approximately 135 ft by 52 ft in size and is 4 stories high with the exhaust air stack extending up another 30 ft above the roof making the total height of the building above the street

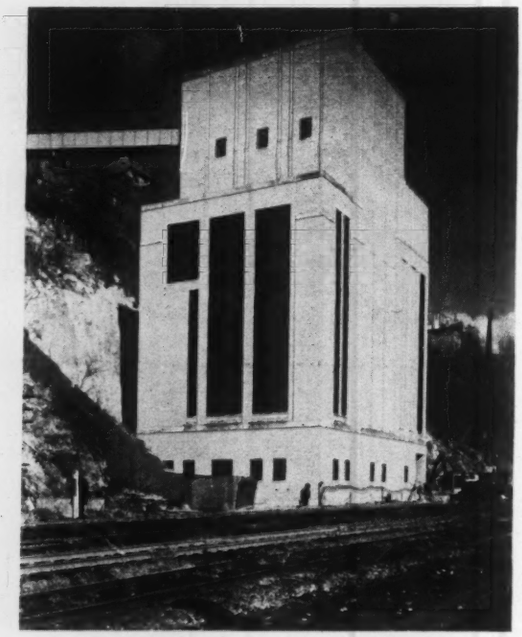


Fig. 5—Exterior view of New Jersey ventilation building

level 116 ft. This building, unlike the other ventilation buildings, was constructed as a complete unit to accommodate the fans for ventilating both the north and south tubes between the river shafts and the portals. The fans for this section of the North Tube have been installed and will remain inoperative until this tube is completed. For the South Tube there are 6 blower fans located on the second, third and fourth floors which draw the fresh air through bronze louvres set in the south and east walls of the building and force it into the fresh air ducts of the tube. The louvres extend almost the full height of the building from about the second floor level to just below the roof. The 6 exhaust fans which are located on the same floors draw the vitiated air from the tunnel ducts and discharge it to the atmosphere through the evasé stacks extending above the roof. The intake louvres are not equipped with motor operated rolling shutters as in the other two buildings since there is no appreciable fire hazard in the surrounding buildings and provision for closing off the intakes was not considered necessary. All fans in this building are chain driven and the motors are set on concrete bases at convenient heights above the floor making them accessible for maintenance and repairs.

The New York River Building is situated in the sli at the foot of West 39th Street. It is built on the con crete filled steel caisson of the river shaft, the east fac being a few feet outside of the U. S. Bulkhead Line The lower portion of the caisson was built in a ship yard, floated to the site and then sunk to bedrock. Th building above, being larger in dimensions than th caisson foundation, is cantilevered out on all sides abov the mean high water level of the river.

This building is about 62 feet 6 in. by 53 ft 4 in. i size and is 6 stories high with the top of the exhaust ai discharge stack parapet 150 ft above the river. The intake louvres are located in the north, south and west walls of the building and extend from just above the third floor to near the roof. As a protection from the smoke of fires that may occur in the neighboring piers all the louvres have motor operated aluminum rolling shutters inside the building so arranged that they can be operated manually also. Each of the 4 exhaust and 3 blower fans is driven by 2-two-speed motors. All fan motors are set on concrete bases above the main floors. The exhaust fan room floors were placed about 5 ft lower than the main corridor floor so that the direct connected motors driving these fans would be at a con venient working height. The New York River Ventilation Building was built as a separate unit to house the fan equipment for the ventilation of the South Tube from this building to the Mid-river bulkhead. When the North Tube is completed there will be another ventilation building identical in size and shape located about 15 ft to the north. The two ventilation buildings will be connected by a cross passage at the tunnel level and also by an enclosed bridge at the sixth floor.

The New Jersey Ventilation Building, illustrated in Fig. 5 and diagrammatically shown in Fig. 6, is located about 1,000 ft inshore from the U. S. Bulkhead Line and approximately 400 ft from the river's edge. It is 100 ft 6 in. by 63 ft 4 in. in size, 7 floors in height and extends to a total height of 150 ft above the ground level, the first floor being some 10 ft above mean high water of the river. It is situated close to the face of Kings Bluff and is connected to the street on top of the bluff by a steel girder foot bridge at the seventh floor level. The louvres are built into the north, south and east walls of the building and extend from about the second floor to within 4 ft of the fifth floor. All the louvres are equipped with the same type of motor operated rolling shutters used in the New York River Building. There are 6 blower and 7 exhaust fans in the building of which three of the blowers and three of the exhausters serve the land portion of the South Tube extending to the New Jersey portal. All the fans for the land section of the tube are chain driven. The 3 blower and 4 exhaust fans for the river section are direct driven by the same combination of 2 motors as in the New York River Building. The motors for all fans are set on concrete bases which are at accessible heights above the floor. The exhaust fans for the river section are in rooms having sunken floors, as in the New York River Building.

The New Jersey Ventilation Building, like the New York River Building, was built as a separate unit for the fan equipment of the South Tube. A new building will be built for the North Tube equipment on new founda-

tions adjacent to the present building so that the two structures will appear as a single unit.

Dampers are provided in each duct between the fan and the common duct leading to the tunnel. These dampers are raised and lowered by means of chains running on sheaves operated by motors with worm gear speed reducers. The control of the damper motor is arranged so that when a fan is started the damper opens and remains open as long as the fan is running. When the fan is stopped the damper closes.

All the exhaust fans are provided with drains in the bottom of their housings to remove the rain water which may enter the discharge stacks.

Fans and Motors

The total number of fans required for the operation of the South Tube is 32 of which 15 are blowers and the other 17 exhausters. One of the fans connected to each duct section is a spare available for use in case of breakdown and is not required to be operated for the maximum ventilation of the tunnel. The fans are all of the double width, double inlet, centrifugal type with backwardly curved blades. The blowers are downward blast discharge and the exhausters upward blast. The housings are of standard steel plate construction and were fabricated in sections to facilitate installation and maintenance. Fan inlets are equipped with vaned inlet cones and electrically welded rigid wire guards. The fans are of standard construction with heavy cast-steel cone hubs and the wheels have 24 blades on each side of the center plate. Shafts of extra stiffness were required by the specification so that the maximum deflection between the bearings would not exceed 0.005 of an inch per foot of length between bearings. Fan bearings are of the roller type, one bearing being fixed and one full-floating for each fan.

Table 1—Blower and Exhaust Fans for South Tube

No. of Fans	System	Duct Section	Location Ventilation Building	Volume Cfm per Fan	Static Pressure In. of Water
3	Blowers	S B 1	New Jersey	106,000	0.500
3	Exhaust	S E 1	New Jersey	108,000	0.375
3	Blower	S B 2	New Jersey	272,000	3.000
4	Exhaust	S E 2	New Jersey	185,000	4.000
3	Blower	S B 3	New York River	280,000	3.750
4	Exhaust	S E 3	New York River	185,000	4.000
3	Blower	S B 4	New York Land	105,000	0.750
3	Exhaust	S E 4	New York Land	108,000	0.375
3	Blower	S B 5	New York Land	105,000	0.750
3	Exhaust	S E 5	New York Land	108,000	1.000

Table 1 shows the number of blower and exhaust fans required for the South Tube together with the section of the tunnel duct to which they are connected, the building in which they are located, the specified volume of air and static pressure when operating at high speed.

Inasmuch as the fans, both blower and exhaust, will be used in various combinations they are of such design that they will operate successfully in parallel. The specifications further called for fans which under any condition of pressure or volume would not cause overloading of the motors and which with a small increase in static pressure would not result in a large decrease in volume of air delivered.

The performance characteristics of the fans are such that high efficiency can be maintained over a wide range of air volume delivery. The maximum horsepower occurs at a point near the point of maximum efficiency and the fans in each case were selected to oper-

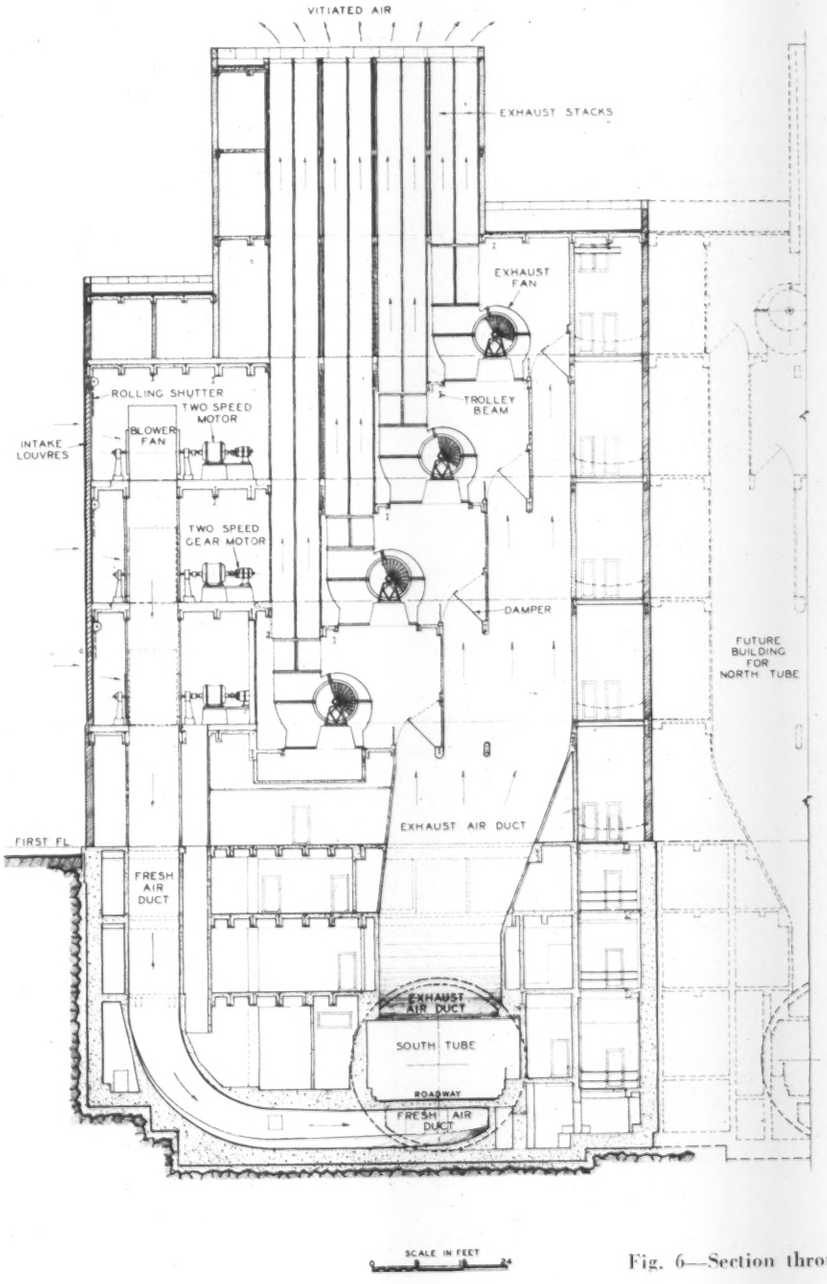


Fig. 6—Section through New Jersey ventilation building

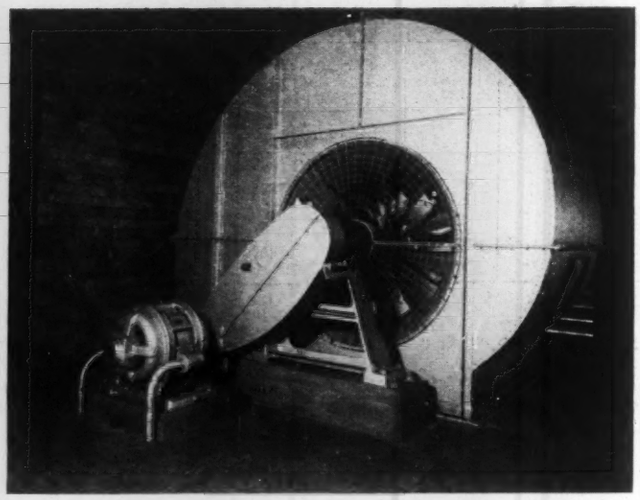


Fig. 7—View of blower fan with chain drive

ate near this point.

In order that fans, motors and transmissions would be of the highest grade and would prove efficient and economical, not only in first costs but also in operating costs, the contract for this equipment required bidders to specify the *Lump Sum Cost* and the *Guaranteed Cost of Operation,* the contract being awarded to the bidder whose sum of these two costs was lowest. The Guaranteed Cost of Operation was determined by the following formula:

$$\text{Cost of Operation (in dollars)} = 0.22 \times H \times 1{,}500$$

in which H is the combined power consumption in kilowatts of all fans, except the spares, when operating at high speed, the factor 1,500 is the probable cost, in dollars reduced to present value, of 1 kw used continuously during the probable life of the equipment. The factor 0.22 is the ratio of the anticipated annual power consumption to the power consumption of all fans, except the spares, if continuously oerated at the high speed. In arriving at the value for H the efficiencies of the drive transmission were taken as 95 per cent for chain drives and 100 per cent for those direct connected. This was done because it was felt that there were not available at that time sufficient data on the efficiency of chain drives to justify the inclusion of that item. Specifying the efficiencies to be used removed this uncertain factor from consideration in comparing bids.

The records of the Holland Tunnel show that the fans, all of which are driven by wound rotor motors, are operated at approximately one-half speed for about 80 per cent of the time and that satisfactory ventilation could be maintained for a considerable part of this time with the fans operating at lower speeds. However, the inherent instability and low efficiency of that type of motor at extremely low speeds made a further reduction impractical. Therefore, in selecting the equipment for the Lincoln Tunnel squirrel cage motors were specified since it was believed that the proper operating speeds of the fans could be determined in advance of actual operation of the tunnel and that by selecting definite motor speeds and types of drive a high efficiency could be maintained at all speeds.

The number of fan speeds and the type of transmission best suited for each duct section were determined by an analysis of the expected traffic load. As the traffic load of the tunnel will be subject to daily and seasonal fluctuations, a large range in speed variation is necessary in order that the volume of air supplied and exhausted can be regulated as the traffic load varies. When the air requirements for each duct section were plotted on a graph against the traffic load for a yearly cycle of operation and evaluated to kilowatt input, it was found that, for the duct section in the land portions of the tunnel, two-speed motors operating at full and one-half speeds would be the most economical in combined initial and operating costs. For the fans in the river sections where the horsepower requirements are considerably higher due to the increased duct lengths, four fan speeds gave the optimum condition as to costs.

The motors for all fans on the land duct sections operate at synchronous speeds of 900 rpm and 450 rpm and drive the fans by means of chain drives enclosed in oil tight cases as illustrated in Fig. 7.

The direct-connected motors driving the fans for the river section ducts shown in Fig. 8, are a combination of two separate motors on one end of the fan shaft. In this arrangement a large two-speed motor is coupled through a flexible coupling to the fan shaft and a smaller geared two-speed motor connected to the extended shaft of the larger motor through a flexible over-running clutch. The large motors run at synchronous speeds of 450 rpm and 300 rpm while the smaller geared unit has output speeds of 225 rpm and 112 rpm. When the larger motor is operating the fan the geared motor is cut out of service by the over-running clutch and when the lower speeds are required the geared motor drives the large motor and the fan. With this drive arrangement low speeds are obtained at high efficiencies and without the use of expensive four-speed motors.

The geared motors are of the single reduction planetary type. All motors for the fans operate on 440 volt, 3 phase, 60 cycle current.

All the fan equipment was painted with a special aluminum paint selected to provide the maximum resistance to the corrosive effect of the salty atmosphere prevalent on the New York water front.

Control System

The electric power for the operation of the tunnel is purchased from public utility companies on both sides of the river, the current being supplied over 3 independent feeders from each company, each of the 6 cables having sufficient capacity to handle the entire tunnel load, thereby providing adequate insurance against power failure. The three 13,200 volt New York feeders are cross-connected and two cables from this source of supply run through the tube to the New Jersey side. The feeders on that side, which are 4,160 volt, are also cross-connected and 2 cables run from that source through the tube to the New York Ventilation Buildings. All the transformers and equipment of the New Jersey power service are suitable for the higher voltage so that if at sometime in the future the voltage should be increased to that in New York it will only be necessary to change some of the connections.

A central supervisory control station is located on the sixth floor of the New Jersey Ventilation Building. The control board located in this room contains the switching

equipment by means of which the operator has complete control over the entire mechanical and electrical equipment of the tunnel. The operator selects the source of power and distributes it to the various units as is desired, operates the fans at the desired speeds and controls the dampers, signal apparatus, and lighting system. Through the telephone system and the code call equipment the attendant can communicate with all the employees in the tunnel or the buildings. Indicators on the control board show the operation of all fans, pumps and auxiliary equipment and the position of the dampers. Potentiometers connected to the carbon monoxide analyzers show and record at all times the amount of this gas in the atmosphere of each of the 5 ventilation sections of the tunnel.

In addition to the central control station there is a local control board in each of the New York Ventilation Buildings, from which the fans in each of these buildings can be operated. Adjacent to each fan there is a local control panel from which that particular fan can be operated. The 3 controls are so interlocked that it is impossible to operate a fan from two points simultaneously.

An interesting feature of the control system is the manner in which the speed of the fan motors is controlled. As all motors are designed for full voltage starting it was necessary to provide some means of preventing the high counter torque which would result if a low speed connection were to be made while the fan and motor were running at a higher speed. The control consists of a small magneto which is gear driven from the end of the fan shaft. As the frequency of the current generated by the magneto varies directly with the speed the circuit can be made to produce a sharp increase in current at critical frequencies by proper adjustment of the capacitance in relation to reactance. Relays responsive to the increase in current cause the motor starting contactors to operate at the proper frequencies which correspond to the selected speed of the motor. The same magnetos also operate the calibrated volt meters which provide the speed indication on the supervisory switchboard.

Heating

The heating of the buildings used in connection with the operation of the Lincoln Tunnel presented no serious problems. The two New York Ventilation Buildings, each of which has only one room that requires heating, have been equipped with electric heaters. The New Jersey Ventilation Building, which contains the supervisory control room, telephone room and toilets for the employees, is heated by a two-pipe steam system and an oil-fired cast iron boiler with full automatic controls. Heat is also provided in a stairway in this building to protect the standpipe from freezing. All radiators are of cast iron, those in the control room being of the concealed convector type. The radiators are equipped with valves of the adjustable orifice type and condensation is returned to the boiler by gravity.

At the New York Plaza near the tunnel portal is the Emergency Garage, a two-story building having a garage on the first floor and offices above. This building has a heating plant which consists of a two-pipe steam system and an oil-fired cast iron boiler. In the garage

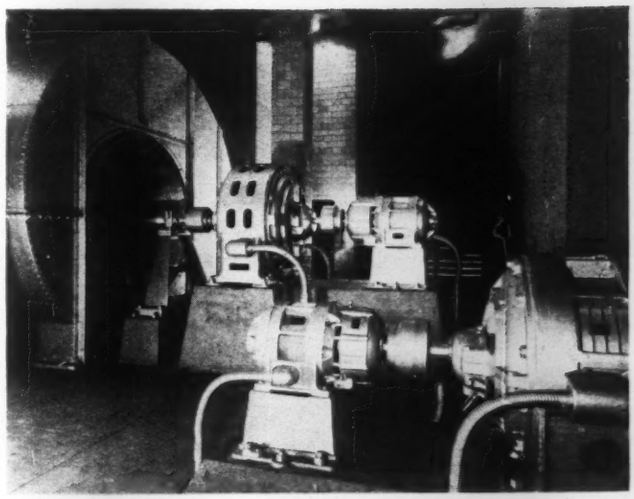


Fig. 8—View of blower fan with direct-connected motors

2 ceiling type unit heaters have been installed. The offices have wall type legless cast iron radiators. The roof of this building has a 1-in. thickness of cork over the concrete roof slab for insulation and the 4 glass skylights have been equipped with heating coils. The control of this system is also fully automatic, the operation of the oil burner being regulated by a thermostat.

The general operating offices of the Tunnel are situated in the Field Office Building adjacent to the New Jersey plaza and provide excellent facilities for the superintendent, the toll collectors, and the Tunnel police. The first floor of this building contains a large garage for the automotive equipment, wrecking cars and fire apparatus, as well as a machine shop for their maintenance. On the second floor are located the general offices, police headquarters, doctors' office, locker rooms, storerooms, toilets, washrooms and other facilities. The building is heated by a two-pipe steam system, with a vacuum return pump and a fully automatic oil-fired cast iron boiler. All radiators in the offices and machine shop are of cast iron, the legless type being used wherever possible. Piping has been run as direct as possible. Float trap drips were used at several points because clearances in the garage necessitated running considerable piping pitched against the steam flow. The garage on the first floor is heated by unit heaters placed near the ceiling. The entire roof is insulated with 1 in. of cork placed over the roof slab.

This building is located on the east side of the plaza directly opposite the line of toll booths which extend across the plaza. In planning the heating system for the building provision was made for heating the toll booths from the same system. An underground pipe gallery extends from the boiler room across the plaza. In the gallery are located the steam and return mains from which connections are provided to the unit heaters in the 14 toll booths.

Because of the various purposes for which the space in the Field Office is used the heating system was zoned to meet the variable heating demands. Four heating zones are used, the toll booths and the garage unit heaters each are on a separate zone and the second floor offices are divided into 2 zones. Each zone has its own

thermostat which operates the oil burner controls and a motorized steam valve on its main near the boiler. Also, the motors of the unit heaters in the garage are operated by the thermostat in that zone. The control system is so arranged that if heat is required in any one of the zones the burner will start and the valve will open to admit steam to that zone without affecting the others.

The heating of toll booths by convection heaters has been unsatisfactory in the past because of the fact that the window being open for a large part of the time, there has been a cold stratum of air at the bottom of the booth while the upper part has been relatively overheated. As a result the toll collectors have suffered from cold legs and feet. In order to overcome this objection it was decided to use a small unit heater with directional louvres in each booth. These heaters have a two-speed manual control. They are located under a shelf at one end of the booth, the center of the heater being 15 in. above the floor.

The booths which are only 6 ft 9 in. long, 2 ft 9 in. wide and 7 ft 4 in. high have glass windows on all four sides. A 1 in. thickness of cork insulation covered by a layer of rubber tile has been provided on the floor as a further aid to the comfort of the collectors. The roof is insulated with rock wool.

L. A. Harding Named Public Works Commissioner of Buffalo

On December 8, Mayor-elect Thomas L. Holling of Buffalo, N. Y., announced the appointment of L. A. Harding, as Commissioner of Public Works. The Mayor-elect said, "Mr. Harding

is one of Buffalo's outstanding engineers, an experienced executive, and a man whose technical ability and personal integrity are of the highest. In keeping with my announced determination to select department heads of outstanding qualifications, I take great pleasure in announcing that Mr. Harding has consented to join my official family and to render a service to the City of Buffalo."

Mr. Harding has resided in Buffalo since 1916 and lives at 85 Cleveland Ave. He graduated with a degree of B. S. and M. E. from Penn State College in 1899 and has had a wide experience in the industrial field as a consultant and author. He was superintendent of Lackawanna Steel Co.'s coal washing plants in Pennsylvania, served as chief engineer for Armstrong Cork Co. and then established his own engineering office.

As a teacher he served as instructor at Cornell University; was Professor of Mechanical Engineering at Penn State and was head of the Department of the Engineering College at the University of Illinois.

For several years he was chief engineer of John Cowper Co. and then organized the L. A. Harding Construction Co. and later the Harding-Carlton Corp., of which he is President. He is also Treasurer of the Leach Steel Corp., Rochester; and the Wilkinson-Harding Corp.

Mr. Harding has been active in association work and has served as president of the *Buffalo Engineering Society*, the *Buffalo General Contractors Association* and the Western New York Air Conditioning Council. In 1930 Mr. Harding was president of the A. S. H. V. E. and has served as Chairman of many important technical committees including the Committee on Research, Committee on Testing and Rating Steam Heating Boilers, Committee on Insulation, Joint A. S. H. V. E.-A. S. R. E. National Standards for Air Conditioning Applications and the Committee on Code of Minimum Requirements for the Heating and Ventilating of Buildings.

He served as a member of the Council from 1928 to 1931 and was President of the Western New York Chapter in 1921. His many friends and associates in the Society will wish him a most successful administration.

Effect of Chemical Treatments on Burning Coal

Chemical mixtures designed for the treatment of coal, with a view to making the coal burn better, seem to have little effect on the combustion of the fuel, according to a recently published report by the engineers of the U. S. Bureau of Mines.

Patent records indicate that the idea of burning fuels by mixing or pretreating them with chemicals dates back to the early years of the last century. In the earlier years the amount of chemical recommended to be used was large, but from 1860 on it has ranged from ½ to 4 lb per ton of coal. Until recent years the sale of such products was restricted to packages sold to householders under the general name of *fuel savers*, with the statement that they would reduce the amount of coal used from 10 to 40 per cent.

Factors studied in this investigation related to the effects of the chemicals on rates of ignition and combustion, on caking of coal in fuel beds, on tar and soot carried from the fuel beds, on soot deposited on surfaces of the furnaces and flues, on the emission of smoke, on the ash and clinker, and on the reaction of the sulphur in the fuels.

Deductions formed from the results of comprehensive tests show that the possible fields of application for the chemical pretreatment of fuels are extremely limited. The effects of light treatments of 4 lb of chemical alone per ton of fuel, disregarding the water that might accompany its use, were so small under normal conditions of burning that they could not account for the improvement that users of treatments in this or smaller amounts have previously believed they observed. The effects are about proportional to the amount of treatment applied.

No efficacy was found for a mixture of chemicals, and each ingredient seemed to act in proportion to the amount that was present. Too much chemical applied to the surfaces of a fuel smothered it and hindered ignition and burning. The tests also showed that any changes that treatments can make in the burning of coal or coke decrease in magnitude as the rate of burning increases, and that differences are small at rates of burning higher than 10 to 15 lb of coal per square foot of grate per hour even with treatments as high as 40 lb per ton of fuel.

Physiological Reactions and Sensations of Pleasantness Under Varying Atmospheric Conditions

By C.-E. A. Winslow,* (*MEMBER*), L. P. Herrington,** and A. P. Gagge*** (*NON-MEMBERS*)
New Haven, Conn.

THE aim of the art of air-conditioning (aside from certain special industrial applications) is to produce conditions which are healthful and pleasant for human beings; and it has been made clear, by the researches of the past quarter-century, that the major effects of the atmospheric environment upon health and comfort are directly related to thermal interchange between the human body and that environment.

Objectives of Study

A comprehension of the nature of these interchanges is essential to sound progress in heating and ventilating practice. Yet the contributions of physiology on this point have been meager. Pettenkofer and Rubner, in the Nineteenth Century, made a good beginning but the leads they furnished have not been followed up, and in the literature of physiology there are practically no fundamental analyses of this problem based on serious experimental attempts to obtain simultaneous measurements of the various factors involved under a wide range of thermal conditions. The most significant contributions have been made by engineers who have boldly and successfully entered the physiological field and among them, the A.S.H.V.E. studies[1] stand preeminent. Largely on the basis of the experiments of these investigators at Pittsburgh, Missenard,[2] an engineer in France, has developed a valuable theoretical analysis of the problem.

The A.S.H.V.E. studies did not permit of the separate evaluation of the influence of radiation. It has been the aim of this work to include this important factor and to attempt an exact analysis of the influence upon physiological reactions and human comfort of widely varying conditions of air temperature, wall temperature, air movement, and humidity, conducted with the most precise physiological instruments available and under conditions which made possible the simultaneous quantitative determination of all the thermal processes involved.

Experimental Methods Employed

For this purpose there has been devised a new method of study which was named *partitional calorimetry* (Winslow, Herrington and Gagge).[3] The subject under observation is placed in a copper booth, from the walls of which radiant heat may be reflected in any desired degree without producing convection currents; the temperature, movement and humidity of the air being controlled by the usual procedures. Metabolism is determined by the ordinary methods. Evaporation from the body is measured directly by weight loss recorded on a delicate scale upon which the subject is seated. Gain or loss of heat by convection is computed from temperature differences between the body surface and the surrounding air; and gain or loss by radiation from temperature differences between the body surface and the surrounding walls. Mean body surface temperature is derived from a weighted average of thermopile observations at 15 points on the body surface. Storage (cooling or heating of the body tissues) is determined from changes in rectal temperature within a range where such determinations are representative; under cold conditions, where the rectal temperature fails to keep pace with cooling of the tissues, storage is obtained indirectly from the difference between the metabolism and the other three factors as directly measured.

Evidence has been presented to show that results obtained in this way and based on the mean of five duplicate experiments have an error of less than 5 per cent (Winslow, Herrington and Gagge[4]).

All of the data discussed in the present paper refer to unclothed subjects in a semi-reclining position. The absolute values cited do not, therefore, apply to ordinary conditions of practice. The fundamental principles, however, must underlie any valid analysis of the relations between the human body and its thermal environment. Studies on clothed subjects are now being carried forward in the laboratory.

Basic Equations and Constants Involved

The fundamental thermodynamic processes concerned may be described by the equation:

$$M + S = E + R + C;$$

where

M = rate of metabolism,
S = rate of storage,
E = rate of evaporative heat loss,
R = rate of radiative heat loss or gain,
C = rate of convective heat loss or gain.

All units are expressed in kilogram-calories per hour,

Contribution No. 21, from the John B. Pierce Laboratory of Hygiene.
*Director, John B. Pierce Laboratory of Hygiene.
**Associate Director, John B. Pierce Laboratory of Hygiene.
***Physicist, John B. Pierce Laboratory of Hygiene.
[1]Heat and Moisture Losses from the Human Body and Their Relation to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. Ed. Miller and W. P. Yant, (A.S.H.V.E. TRANSACTIONS, 1929, Vol. 35, p. 245).
[2]Etude Physiologique et Technique de la Ventilation, by F.-A. Missenard, L. Eyrolles, Paris, 1933).
For presentation at the 44th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at the Joint Session with the *A.S.R.E.*, New York, N. Y., January, 1938.
[3]A New Method of Partitional Calorimetry, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge, (*American Journal Physiology*, 1936, Vol. 116, p. 641).
[4]The Determination of Radiation and Convection Changes by Partitional Calorimetry, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge, (*American Journal Physiology*, 1936, Vol. 116, p. 669).

and storage is considered positive when the body cools, negative when the body becomes warmer. R and C are positive when T_S is greater than T_W and T_A, respectively; negative when T_S is less than T_W and T_A.

Metabolism and evaporative heat loss are determined by direct experimental methods. Storage, as pointed out heretofore, is either calculated from the rate of change in body temperature or computed by difference.

Radiation interchange is governed by the Stefan-Boltzmann Law, and expressed by the equation,

$$R = A_R\, k\, (T_W^4 - T_S^4);$$

or

$$= A_R\, (4k\, T_S^3)\, (T_W - T_S), \text{ as a first approximation,}$$

where

R = radiation exchange,
A_R = effective radiation area of the body in square meters,
k = the universal radiation constant,
T_W = mean black body radiant temperature of environing surface in absolute degrees,
T_S = mean skin temperature in absolute degrees.

The unknown factor here is A_R, the effective radiation surface of the body. This will, of course, be less than the total surface of the body and will vary with body build and position during the experiment. It has been necessary, therefore, to determine this factor for each individual, which can readily be done by holding the difference between skin and air temperature constant (which holds convection loss also constant) and varying wall temperatures (Gagge[5]).

Convection loss will depend upon mean skin temperature, air temperature, and air movement. The constants involved in convection loss vary with the form of the object cooled and even the type of formula which applies to the phenomenon is not agreed upon.

The formula most commonly used is as follows:

$$C = A\, (\triangle T)^a + B\, (\triangle T)\, V^b,$$

where $\triangle T$ represents the difference between skin and air temperature, and *where* A, B, a and b, are constants. For such limited air velocities as occur under ordinary conditions of ventilation, the exponential constants may be considered as equaling unity. The formula then becomes:

$$C = (A + B\, V)\, \triangle T.$$

Other students of the problem have employed a quite different relation:

$$C = A\, \sqrt{V}\, \triangle T.$$

Here there is but one arbitrary constant and convection loss is assumed to vary with the square root of air velocity.

The authors' studies[6] indicate that either of these two formulae will describe observed results with equal accuracy. If the entire curve is parabolic, the authors' experiments are obviously in a section of the curve which is essentially linear. Therefore, the square root type of formula has been used as equally accurate and simpler of application, and A has been determined for each subject at varying air temperatures.

The fundamental partitional equation, then, assumes the following form:

$$M + S = E + A_R K\, (\triangle T_W) + A_C\, (\triangle T_A)\, \sqrt{V},$$

where

K = $(4kT_S^3)$ from the first approximation of Stefan Law,[7]
A_R = the effective radiation area for a given subject,
$\triangle T_W$ = difference between skin and wall temperatures,
A_C = convection constant for a given subject,
$\triangle T_A$ = difference between air and skin temperatures,
V = air velocity.

General Relations Observed

To simplify analysis of the results obtained first consideration shall be given the effects of radiation and convection as combined, and the thermal interchanges shall be partitioned under the four headings of metabolism, evaporation, storage, and radiation plus convection. The distinctive influence of radiant heat will be examined in a later paragraph. For such an analysis, the figures must obviously be compared for energy interchange with a temperature scale which represents the combined influence of both radiation and convection; and for this purpose there has been introduced the concept of *operative temperature*. *Operative temperature* is the temperature which represents the net physical effect of a given combination of air and wall temperatures (Winslow, Herrington and Gagge[8]; Gagge, Herrington and Winslow[9]).

The general results obtained with two subjects are presented in Fig. 1. Subject I was an extremely stout individual and Subject II a very slender one, representing the extremes of ordinary body build. The velocity of turbulent air movement in the experiments plotted here was 15-20 linear feet per minute, and relative humidity between 40 and 50 per cent. It will be noted that operative temperatures vary from 64 to 108 F. These figures represent the resultant of much more widely ranging air and wall temperatures. Actually, air temperatures varied from 43 to 97 F and wall temperatures from 66 to 135 F. In most experiments the walls were warmer than the skin and the effect of radiation ranged from a loss of 84 to a gain of 189 kilogram-calories per hour. The air was generally cooler than the skin and the effect of convection ranged from a loss of 183 to a gain of 4 kilogram-calories per hour. The following phenomena are obviously apparent in both graphs.

1. Metabolism (for a given subject) remained approximately constant within the range of operative temperatures employed. This would not, of course, be the case under more extreme conditions.

2. At a critical temperature of 88 to 90 F the heat produced by metabolism (roughly 100 kilogram-calories per hour for Subject I) is balanced by the heat loss due to evaporation and to radiation plus convection—these two major components accounting for about 50 kilogram-calories each. Storage is zero; that is, the body tissues show no change in temperature.

3. As one proceeds to higher operative temperatures, the heat received from the environment due to radiation plus convection increases progressively but is exactly balanced by a similarly progressive increase in evaporative heat loss, so that no negative storage (warming of the body) takes place. At an operative temperature of 106 F the body is gaining 100 kilogram-calories from the combined influence of walls and air, and this heat gain plus the metabolic heat produced is balanced by a heat loss of 200 kilogram-calories due to evaporation.

4. Below the critical temperature of 86 F the phenomena are

[5]The Linearity Criterion as Applied to Partitional Calorimetry, by A. P. Gagge, (*American Journal Physiology*, 1936, Vol. 116, p. 656).
[6]Thermal Interchanges Between the Human Body and Its Atmospheric Environment, by A. P. Gagge, L. P. Herrington and C.-E. A. Winslow, (*American Journal of Hygiene*, 1937, Vol 26, p. 84).
[7]For purposes of analysis, K may be evaluated as a constant by assuming a mean value of T_S over the range of experiment.
[8]Physiological Reactions of the Human Body to Varying Environmental Temperatures, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge, (*American Journal Physiology*, 1937, Vol. 120, p. 1).
[9]Loc. Cit. See Note 6.

wholly different. Evaporative heat loss here changes but slightly, falling from 40 to 20 kilogram-calories per hour as the operative temperature decreases, as a result chiefly of the purely physical factor of decreasing vapor pressure difference between skin and air. In this zone, heat loss due to the combined influence of radiation plus convection increases progressively (although the slope of the line is less rapid than above 86 F for reasons to be discussed in a later paragraph). Since this progressively increasing heat loss is not balanced, there is a parallel increase in storage (cooling of the body tissues).

It is clear that no simple equation can describe the influence of atmospheric conditions above and below the critical point in these graphs since the relative influence of air temperature, air movement and relative humidity will be quite different on the two sides of the inflection point. The area above 88 F has been called the Zone of Evaporative Regulation, and the area below 86 F the Zone of Body Cooling, and the phenomena apparent in each of these zones must be considered separately.

Reactions in Zone of Evaporative Regulation

In this area, it has been observed that the effects of an increasingly warm environment are exactly balanced by a parallel increase in evaporative cooling. At low operative temperatures (below 81 F) a constant evaporative loss of about 30 kilogram-calories per hour is observed which represents minimum evaporation from the skin in the absence of sweat secretion. As operative temperature rises in the Zone of Evaporative Regulation, skin temperature also rises and when the skin temperature reaches 93 F active sweat secretion begins. Above this point skin temperature remains approximately constant but with progressively rising operative temperature sweat secretion increases in exactly parallel fashion (Winslow, Herrington and Gagge[10]). To express the effect of the increase in sweat secretion there has been derived a new physiological constant *wetted area* which describes the extent of moisture surface present on the total surface of the body (Gagge[11]; and Winslow, Herrington and Gagge[12]).

Influence of Relative Humidity

The experiments plotted in Fig. 1 were conducted with a constant relative humidity of 40 to 50 per cent. In other studies this factor has been varied (Winslow, Herrington and Gagge[13]). It has been found that the body can balance changes in relative humidity within this zone just as precisely as it can balance changes in temperatures —up to the point where the sweat no longer evaporates but runs off without exerting its cooling power. Until the latter point is reached, evaporative heat loss at a given air temperature is exactly the same, whether humidity be high or low. As the vapor pressure of the atmosphere increases, the wetted area increases *pari passu* so as to maintain the heat loss which is essential.

In Fig. 2 are presented curves indicating how wetted area changes in this zone with varying air temperature and relative humidity (the walls being at the same temperature as the air in all cases). The 10 per cent line (wetted area equal to 10 per cent of the potentially wettable area) corresponds to the minimum evaporation occurring without active sweat secretion. The 100 per cent line corresponds to the condition beyond which sweat no longer evaporates but collects and runs off the body without exerting its cooling power. Above the points on this line the process of evaporative cooling fails and the body temperature must rise.

Influence of Air Movement

The discussion in preceding paragraphs, and the data presented in Fig. 2 refer to conditions in which there was a turbulent air movement of 17 linear feet per minute. With variation in air movement, the rate of evaporation and the wetted area will change in accordance with formulae which have been

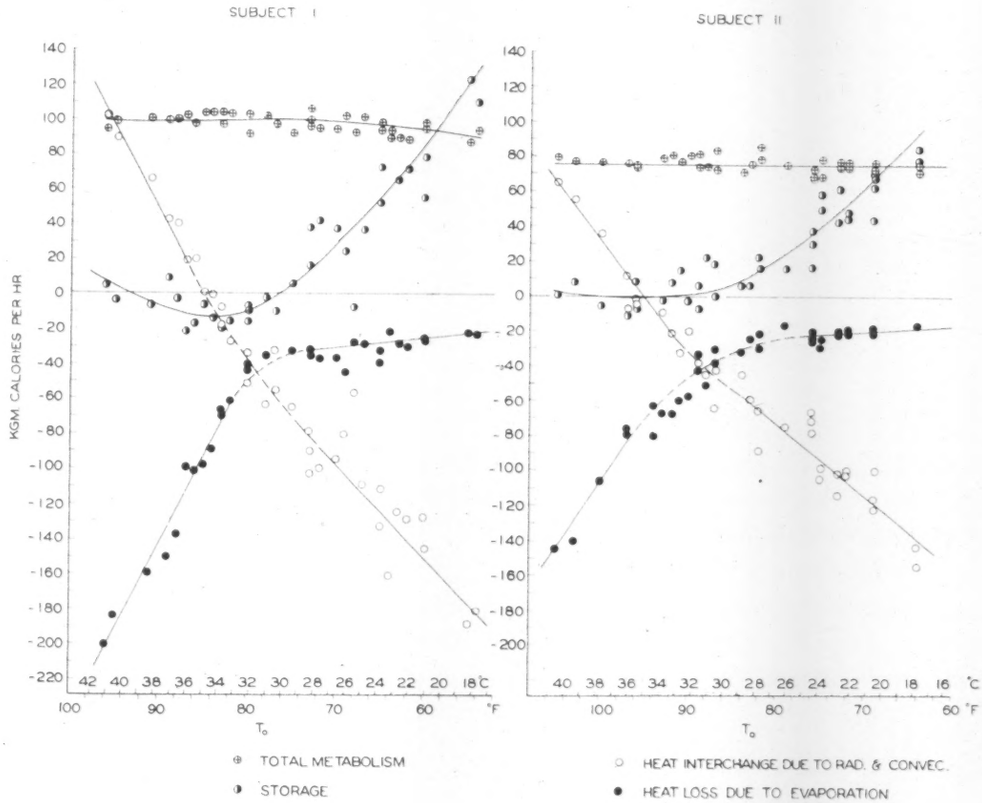


Fig. 1—Elements of heat partition for a stout (I) and a slender (II) subject at varying operative temperatures (T_o)

[10]Loc. Cit. See Note 8.
[11]A New Physiological Variable Associated with Sensible and Insensible Perspiration, by A. P. Gagge, (*American Journal Physiology*, 1937, Vol. 120, p. 277).
[12]Physiological Reactions of the Human Body to Various Atmospheric Humidities, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge, (*American Journal Physiology*, 1937, Vol. 120, p. 288).
[13]Loc. Cit. See Note 12.

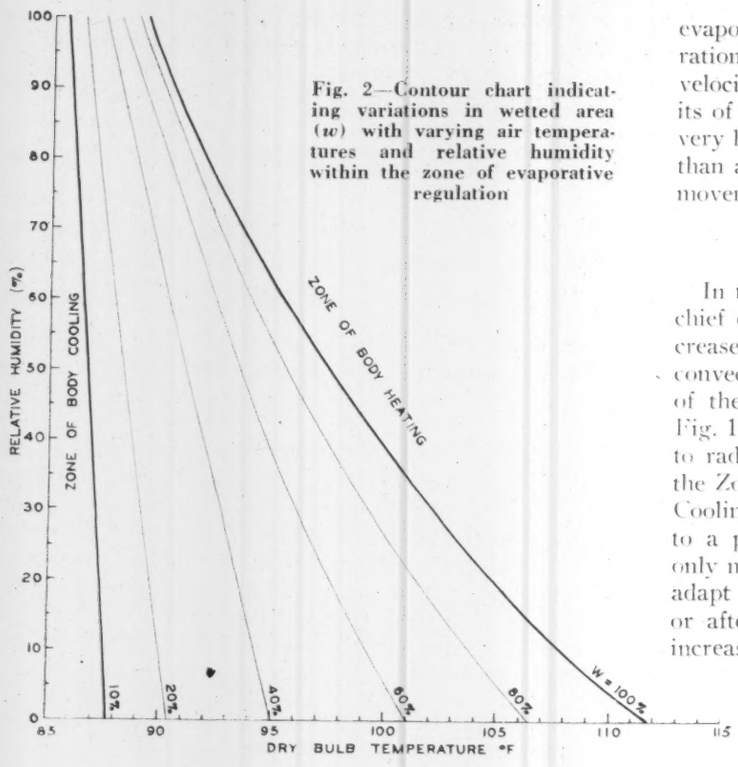


Fig. 2—Contour chart indicating variations in wetted area (w) with varying air temperatures and relative humidity within the zone of evaporative regulation

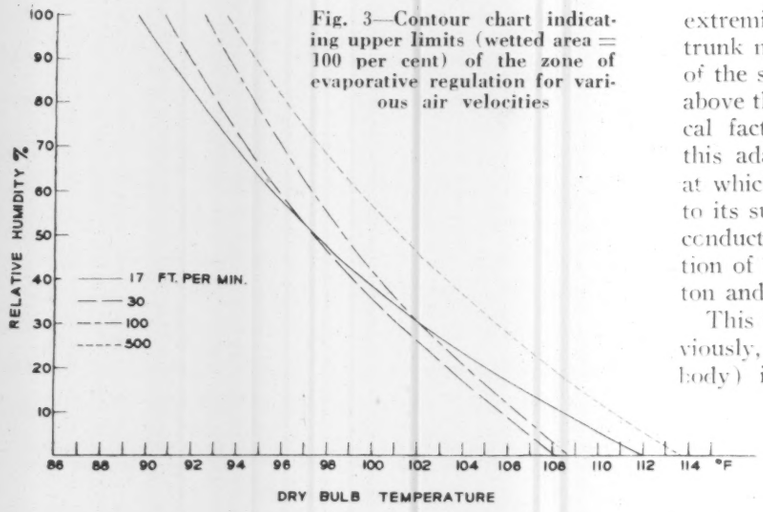


Fig. 3—Contour chart indicating upper limits (wetted area = 100 per cent) of the zone of evaporative regulation for various air velocities

presented elsewhere (Gagge, Herrington and Winslow[14]). The effect of this factor can best be understood by analyzing its influence upon the 100 per cent line of Fig. 2, representing the upper limits of possible evaporative regulation. This influence is illustrated in Fig. 3.

It will be noted that (for the nude subject in a semi-reclining posture) increase in air movement consistently increases evaporative cooling, and therefore heat tolerance, when relative humidity is high and air temperature low. When relative humidity is low and air temperature is high, however, an increase in air velocity from 17 to 30 or 100 linear feet per minute actually decreases heat tolerance. Under these conditions air temperature is above skin temperature and the increased demand for evaporative cooling exceeds the actual increase of evaporation due to the higher air movement. When an air velocity of 500 linear feet per minute is reached, the limits of the zone are broadened throughout its range. For very hot and dry environments, still air is more desirable than a slightly greater air movement but a very high air movement is still better.

[14]Loc. Cit. See Note 6.

Reactions in Zone of Body Cooling

In this area (below 86 F for the unclothed body), the chief changes which occur, as operative temperature decreases, are (a) increased heat loss due to radiation plus convection, and (b) increased positive storage or cooling of the body tissues. It will be noted by reference to Fig. 1 that the slope of the line representing heat loss due to radiation plus convection changes as one passes from the Zone of Evaporative Regulation to the Zone of Body Cooling. The less abrupt slope in the latter zone is due to a progressive fall in skin temperature which is the only mechanism the body calls into play in this region to adapt to a cool environment. Under colder conditions, or after longer periods of time, a second mechanism—increased metabolism—would become operative but this does not appear in the experiments here reviewed.

For a fall in operative temperature from 88 to 68 F the mean skin temperature decreases from 94 to 84 F. The temperature of the lower extremities falls most rapidly while that of the head or trunk may decrease less. In general, however, all areas of the skin tend to maintain a differential of at least 8 F above the environment. There has been derived a physical factor, conductance, which measures the result of this adaptive variation, expressed in terms of the rate at which heat passes from the deeper tissues of the body to its surface. It depends on a constant factor of tissue conductivity for the individual and on the state of dilation of the superficial blood vessels (Winslow, Herrington and Gagge[15]).

This type of regulation is, however, as pointed out previously, incomplete; and positive storage (cooling of the body) increases progressively as air temperature falls. Fig. 4 shows the effect upon skin temperature of various combinations of air and wall temperature for the unclothed body (Gagge, Herrington and Winslow[16]).

Influence of Air Movement

It is, of course, obvious that air movement will always increase heat loss due to convection, while evaporation (which is quantitatively unimportant in this zone), will not be greatly influenced. Radiation will not be directly affected but radiative heat loss to cold walls will be decreased by the fall of skin temperature which results from greater convective loss, and radiative heat gain from hot walls correspondingly increased. The relation of storage to air and wall temperatures for various air velocities has been derived as follows:

For a velocity of

17 fpm; $S = 232.2 - 1.03\ T_A - 1.60\ T_W$,
28 fpm; $S = 246.0 - 1.22\ T_A - 1.55\ T_W$,
50 fpm; $S = 276.0 - 1.60\ T_A - 1.50\ T_W$,
100 fpm; $S = 290.0 - 1.74\ T_A - 1.45\ T_W$,

[15]Loc. Cit. See Note 8.
[16]Loc. Cit. See Note 6.

where S equals storage in kilogram-calories per square meter of body surface per hour.

It should be made clear that storage, as defined by the authors, is a measure of the deviation of the body from thermal equilibrium after a specified time of exposure (in these experiments, one-and-a-half hours). It does not measure the total heat loss during this period but the rate of heat loss at the end of it.

Radiation as Distinguished from Convection

So far, heat losses and heat gains due to radiation and convection have been considered, as if their influence upon the body corresponded entirely to their potency as measured by ordinary physical instruments. This is the case in the Zone of Evaporative Regulation (where no significant changes in skin temperature are manifest). In the Zone of Body Cooling, however, distinctly different physiological reactions to the two forms of heat interchange may be observed (Herrington; Winslow, and Gagge[17]).

If two conditions are compared in which the same operative temperature is maintained—by warm walls and warm air and by hot walls and cold air respectively—markedly different responses are found in skin temperature (and hence in conductivity of the superficial body tissues). With cold air and hot walls the skin temperature falls more rapidly than with air and walls, of more nearly equal temperature, exerting the same total thermal effect from a physical standpoint.

If metabolism and evaporation remain normal, there is found the following prediction equation, indicating the greater influence of air than wall upon skin temperature:

$$T_S = 0.29\ T_A + 0.15\ T_W + 56.5.$$

Whether this differential effect is due to a specifically stimulating effect of cold air, to a specifically stimulating effect of hot radiation, or to both combined, is not known. It is clear, however, that with warmer walls and cooler air the body adapts itself more readily to cold conditions than it does to a thermally-equivalent environment in which air and walls are at the same temperature.

Sensations of Pleasantness in Zone of Evaporative Regulation

In connection with the studies here discussed, the subjects were asked at the close of each experiment to record their subjective impressions with regard to the atmospheric conditions maintained, on a scale of five points, as follows: (1) very pleasant; (2) pleasant; (3) indifferent; (4) unpleasant; and (5) very unpleasant. These expressions of opinion proved to be correlated with physical conditions and physiological reactions to an even higher degree than had been anticipated (Winslow, Herrington and Gagge[18]).

At the point of thermal equilibrium (operative temperature between 88 and 90 F) there were eight groups of experiments. In five of these groups, the vote was 2 (pleasant) or less; in two groups it was between 2 and 2.5; in one group it was 3 (indifferent). This is the range in which skin temperature is normal (88-93 F), storage minimal and evaporation low. On either side of this region votes became progressively higher (more unpleasant) when proceeding to hotter or colder conditions.

On the hot side, in the Zone of Evaporative Regulation, significant correlations were found between sensations of pleasantness on the one side and both skin temperature and wetted area on the other. When wetted area is held statistically constant, however, the correlation with skin temperature becomes insignificant; while, when skin temperature is held constant the correlation between sensation and wetted area remains $+0.51 \pm 0.08$. It is found that, within this zone the relation of votes as to pleasantness may be expressed by the prediction equation:

$$C = 0.022\ w + 1.95,$$

where

C = sensations of comfort or pleasantness, with high value indicating unpleasantness.

w = wetted area.

The relations are graphically indicated in the left half of Fig. 5.

In this Zone, at a given rate of air movement, wetted area (and hence the sensation of pleasantness) is determined by relative humidity and operative temperature (or air temperature if air and wall temperature are the same). The range of sensation votes for the latter condition, with respect to air temperature and relative

[17]The Relative Influence of Radiation and Convection Upon Vasomotor Temperature Regulation, by L. P. Herrington, C.-E. A. Winslow and A. P. Gagge), (*American Journal Physiology*, 1937, Vol. 120, p. 133).

[18]Relations Between Atmospheric Conditions, Physiological Reactions and Sensations of Pleasantness, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge, (*American Journal of Hygiene*, 1937c, Vol. 26, p. 103).

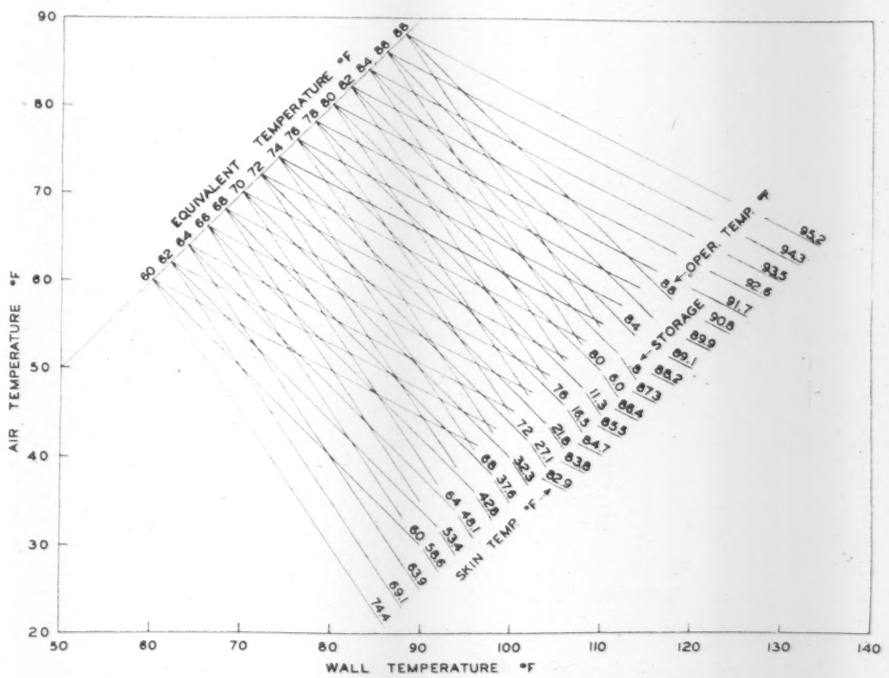


Fig. 4—Contour chart for prediction of skin temperature and storage for various combinations of air temperature and wall temperature (relative humidity 40-50 per cent; air movement, 17 fpm)

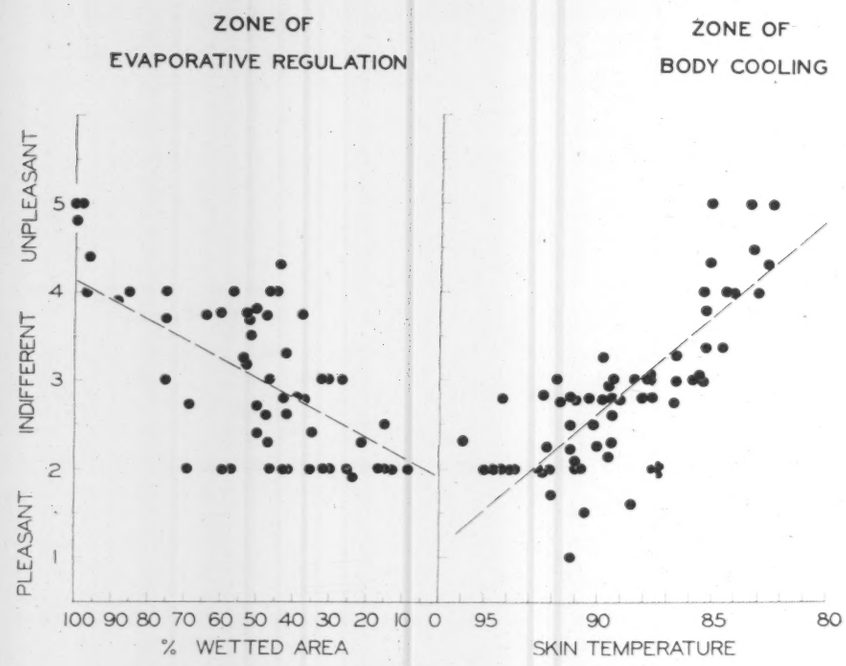


Fig. 5—Relation of sensations of pleasantness to wetted area (in the zone of evaporative regulation) and to skin temperature (in the zone of body cooling)

humidity, is indicated in Fig. 6. This graph is based on an air movement of 17 linear feet per minute. Higher air movement will modify the effects as noted in an earlier paragraph.

It is interesting to note that throughout the Zone included between the lines of Fig. 6, thermal regulation is perfect. Sweat secretion is successfully balancing the warmth of the environment; but this process—in spite of the fact that sweat is evaporating as fast as it is formed—is accompanied by marked sensations of unpleasantness. For an air movement of 17 linear feet per minute, the degree of sweat secretion associated with sensations of distinct unpleasantness will occur beyond the following limits of temperature and relative humidity.

Temperature, F	Relative Humidity, Per Cent
89	96
92	70
95	48
98	30
101	19
104	0

Sensations of Pleasantness in Zone of Body Cooling

In this region widely different phenomena are observed. Here, there is a small correlation between sensations of pleasantness and storage, which is not statistically significant, and a highly significant correlation between sensations of pleasantness and skin temperature (-0.64 ± 0.09). The general relationship is indicated on the right half of Fig. 5.

It is found that, in this area, the pleasantness of the thermal environment may be predicted from the relation:

$$C = (-0.22)\, T_S + 22.4.$$

The chief variables affecting skin temperature (and hence sensation) in this Zone are operative temperature (or air temperature, if air and wall temperatures are the same) and air movement. The limits for various degrees of pleasantness or unpleasantness are indicated in Fig. 7.

Where air and wall temperatures differ, it has been pointed out that the influences upon skin temperature of radiant heat and convected heat are different. With cold air and warm walls, skin temperature will be lower and discomfort greater than with the same operative temperature produced by walls and air of identical temperature.

The following conditions represent limits beyond which distinct sensations of unpleasantness will occur with different combinations of air and wall temperature and with air velocities of 17 and 100 linear feet per minute:

Air Movement 17 fpm		Air Movement 100 fpm	
Air Temp. F	Wall Temp. F	Air Temp. F	Wall Temp. F
68	68	76	76
65	73	73	91
62	80	70	105
59	86	67	118

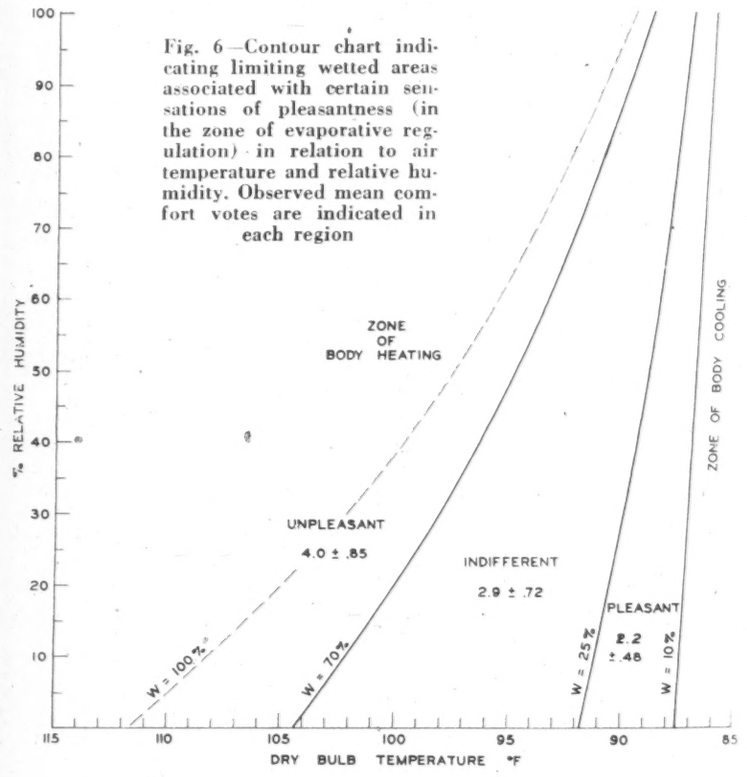


Fig. 6—Contour chart indicating limiting wetted areas associated with certain sensations of pleasantness (in the zone of evaporative regulation) in relation to air temperature and relative humidity. Observed mean comfort votes are indicated in each region

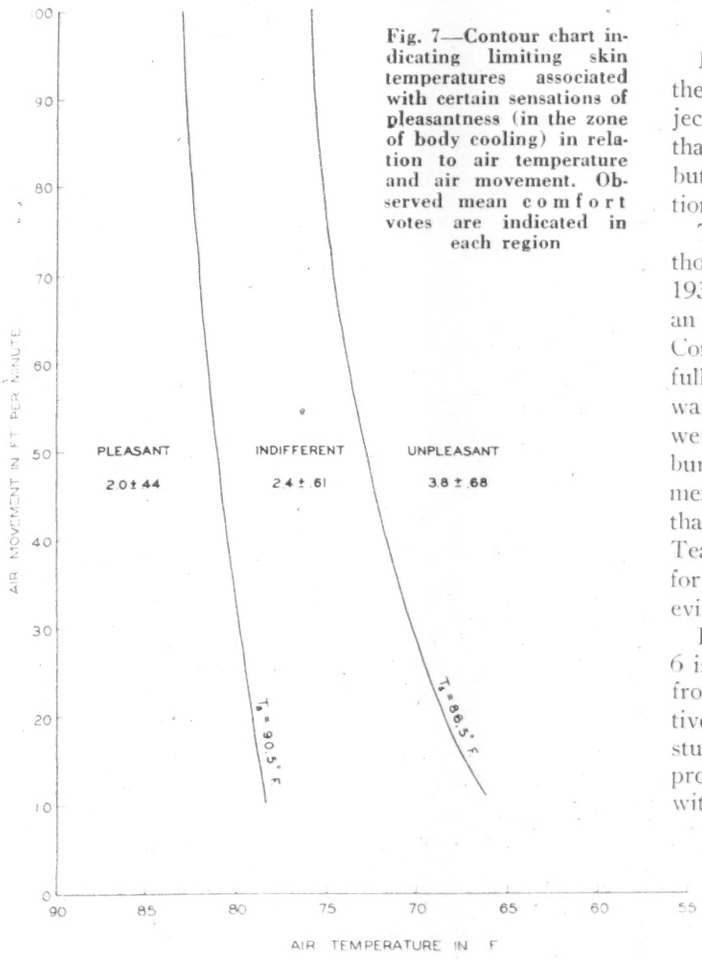


Fig. 7—Contour chart indicating limiting skin temperatures associated with certain sensations of pleasantness (in the zone of body cooling) in relation to air temperature and air movement. Observed mean comfort votes are indicated in each region

Limitations of the Present Study

It cannot be too clearly emphasized that the results of the studies here reported refer only to the unclothed subject, at rest in a semi-reclining position. It is believed that the principles developed are of general application but the absolute values derived apply only to the conditions of the experiments.

These data cannot, therefore, be directly compared with those of the A.S.H.V.E. Comfort Chart (THE GUIDE, 1937). These subjects were always unclothed except for an athletic supporter, while the Pittsburgh subjects (McConnell, Houghten[19]; Houghten and Yagloglou[20]) were fully clothed under cold conditions, and stripped to the waist only, under hot conditions. The present subjects were at rest in a semi-reclining position while the Pittsburgh subjects were allowed a certain freedom of movement. Metabolisms were, therefore, substantially higher than those in the tests herein discussed (Houghten, Teague, Miller and Yant[21]) so that limiting temperatures for unpleasant conditions on the cold side were also inevitably higher than those of the present authors.

It is a little puzzling that the slope of the lines in Fig. 6 is substantially different from what would be expected from the lines of the Comfort Chart, the influence of relative humidity seeming to be less marked in present studies. Any detailed analysis of such differences can profitably be postponed until the work now being done with clothed subjects is complete.

[19]Some Physiological Reactions to High Temperatures and Humidities, by W. J. McConnell and F. C. Houghten, (A.S.H.V.E. TRANSACTIONS, 1923, Vol. 29, p. 129).
[20]Determination of the Comfort Zone, by F. C. Houghten and C. P. Yagloglou, (A.S.H.V.E. TRANSACTIONS, 1923, Vol. 29, p. 361).
[21]Loc. Cit. See Note 1.

Sterilization By Radiation

The purification of the air we breathe by the use of a sterilizing lamp, was predicted in a recent talk by A. R. Dennington, at the Canadian Section, *A.I.E.E.*, in order to combat the contamination of air impurities resulting from the concentration of many people in urban centers.

Since the discovery of the existence of micro-organisms, attempts have been made to control them, by creating environments favorable to those which it is desired to have survive. Attempts to reduce bacterial growth are usually referred to as sterilization, although it should be realized that complete sterility is an ideal that cannot be obtained.

Sunlight is considered as a germicide and is effective even though both ultra-violet and infra red rays are filtered out. Mr. Dennington stated that various experimenters have demonstrated that the shorter wave lengths are more effective than the longer ones in their lethal action. For a given expenditure of radiant energy, the most effective radiation is in the region of 2537 angstroms and down to 2000 angstroms or even less. The extremely short wave lengths are readily absorbed in the air and hence are effective only at short distances from the source.

For sterilizing problems, a tubular lamp is available made of special glass and containing low pressure mercury vapor as well as a mixture of inert gases, such as argon and neon, to reduce the starting voltage. The lamp is tubular in shape to keep the path of the rays within the lamp itself as short as possible as the mercury vapor absorbs the radiations and if in large volume reduces the useful output. The temperature of the tube is only a few degrees above that of the surrounding air so that it may be installed in places where any marked increase in the heat load is a disadvantage.

Sterilizing lamps have been adapted to materially reduce the hazards of surgical operations, according to Mr. Dennington, and there is a possibility that this procedure may become standardized hospital practice. It was also stated that perhaps, the efficiency of office workers may be increased by not only having the air properly heated or cooled and humidified to meet health requirements, but also treated to reduce the bacteria count, thus, releasing additional energy for battling the unseen microscopic enemies.

Correlating Thermal Research

As a part of the efforts of the A. S. H. V. E. Research Laboratory to correlate research in thermal engineering carried on by the many institutions engaged in such work, and to disseminate the published results of such studies together with other reports of progress in the field, and in order to make this information available to the membership of the Society, there will be published monthly on this page a limited number of brief abstracts of articles which it is believed will be of interest to all concerned. For more complete lists address the Librarian, A. S. H. V. E. Research Laboratory, U. S. Bureau of Mines Experiment Station, Pittsburgh, Pa.

F. C. Houghten, *Director*.

W. A. Danielson, *Chairman*,
COMMITTEE ON RESEARCH

• Modern Methods of Measuring and Predetermining Daylight, by Percy J. Waldram. *Journal of the Royal Institute of British Architects*, vol. 44, 3rd series, no. 20, Oct. 16, 1937, p. 1045. Discussion of daylight ratio of 0.4 per cent sill ratio or 0.2 per cent daylight factor as the grumble point for any location in ordinary rooms. Possibility of defining criteria of amenity for natural illumination. Application of such criteria to all buildings. Possibility of expressing by-laws to maintain criteria of illumination comfort.

• The History of Building Regulations, by George N. Thompson. *Building Standards Monthly*, Oct. 1937. See also abstract in *Industrial Standardization and Commercial Standards Monthly*, vol. 8, no. 11, Nov. 1937, p. 291. Discussion of regulations for construction of buildings from early times up to present date. Early Egyptians and Babylonians had strict building regulations. Building contractor was held responsible for strength and safety of his buildings. Department of Commerce Building Code Committee worked 13 years with the National Bureau of Standards; eight reports were published on building regulations.

• Air Conditioning, by Erick Larson. *American Gas Journal*, vol. 147, no. 3, Sept. 1937, pp. 35-40. General discussion of air conditioning, its value in maintaining comfort and health of man.

• Air Conditioning With a Lithium Chloride System, by S. R. Knapp. *Cas*, vol. 13, pp. 20-24, 1937. Description of a lithium chloride absorption system adapted for air conditioning an auditorium used for short periods. Lithium chloride sprayed over baskets of glass wool.

• Air Conditioning With Well Water and Refrigeration. *Domestic Engineering*, vol. 150, no. 4, Oct. 1937, p. 72. Description of installation of heating and cooling in a four-story office building in Louisville, Ky. Roof sprinkler system aids cooling. Increased rentals noted.

• A Study of Office Air Conditioning Using Ice, by Clifford F. Holske. *Heating and Ventilating*, vol. 34, no. 11, Nov. 1937, pp. 31-34. Description of ice cooled office air conditioning plant on 21st floor of a New York office building. Diagram of ice bunker and duct layout for the office.

• Flanged Pipe Joints for High Pressures and Temperatures, by R. W. Bailey, D.Sc. *Engineering*, vol. CXLIV, no. 3742, Oct. 1, 1937, pp. 364-365, continued in several numbers. Part 1. elastic behavior, part 2. creep behavior and application of the results of analysis to practice. This analysis gives the stress distribution and deformation of a heavy ring of rectangular cross-section and extends the treatment to include the effect of bolt holes, of the fillet between the flange and pipe and of pipe wall, upon deflection at the flange.

• A Study of Dust Control Methods in an Asbestos Fabricating Plant, by Richard T. Page and J. J. Bloomfield. *Public Health Reports by the U. S. Public Health Service*, vol. 52, no. 48, Nov. 26, 1937. An extensive medical and engineering study of the health of asbestos workers has been conducted by the U. S. Public Health Service. This paper supplements the general study with a detailed study of the dust control methods used in an asbestos fabricating plant.

• Refractory and Insulating Materials for Small Heating Plants, by Spotts J. McDowell. *Automatic Heat and Air Conditioning*, vol. 8, no. 11, Nov. 1937, pp. 54-57. Discussion of proper selection and wise application of refractory and insulating materials.

• Duct Design and Layout. *Automatic Heat and Air Conditioning*, vol. 8, no. 11, Nov. 1937, pp. 45-47. Detailed description of sheet metal duct construction for air conditioning systems. Diagrams of seams and joints, also method of cutting and shaping a duct from a piece of sheet metal.

• Friction and Heat Transmission in Fluids. *Engineering*, vol. CXLIV, no. 3749, Nov. 19, 1937, p. 573. Discussion of paper, Friction and Heat Transmission Coefficients, by W. F. Cope, published by *Institution of Mechanical Engineers* for discussion in writing. Paper deals with experimental work upon water flowing in pipes of circular and various rectangular cross-sections in which the frictional and heat transmission coefficients were measured simultaneously. Heat transmission for a rectangular pipe can be estimated by comparison with a circular pipe on basis of hydraulic diameter. In this respect the heat transmission coefficient is analogous to the frictional resistance. Also discussion of roughened pipes.

• Physiological Reactions of the Human Body to Various Atmospheric Humidities, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge. *The American Journal of Physiology*, vol. 120, no. 2, Oct. 1937, pp. 288-299. Report of tests at John B. Pierce Lab. of Hygiene, New Haven, Conn. Physiological reactions of three unclothed male subjects in atmosphere of uniform turbulent air, no artificial radiation; air temperature and relative humidity were variables.

• New Research Program at Pittsburgh Plate. *The Glass Industry*, vol. 18, no. 12, Dec. 1937, pp. 411-414. Description of organization of the Central Research Department of Pittsburgh Plate Glass Co. personnel and laboratory. Purpose of the laboratory and some of the results of past two years' work.

• Heating, Ventilating and Air Conditioning Fundamentals, by William H. Severns, University of Illinois. Published by John Wiley & Sons, Inc., 440 Fourth Ave., New York City. 467 pages, 284 illustrations, price $4.00. Presentation of essential fundamentals of the subject usually offered in courses for undergraduate students in Architecture, Architectural Engineering and Mechanical Engineering. A full discussion of applicable theory and typical examples of calculations.

• The Geometry of Conical Pipes, Bends and Joints, by W. Sellar. Pamphlet published by the Draughtsman Publishing Co., Ltd. Price 2s net. Many drawings. Method of making neat and practicable bend connections in sheet metal material between one cylindrical pipe or duct and another. The geometrical solution of the problem makes use of fact that the elipse is a plane section both of the cone and of the cylinder. Basic principle of the method of construction is explained. No geometrical proof given. Drawings are sufficient guide to a draftsman.

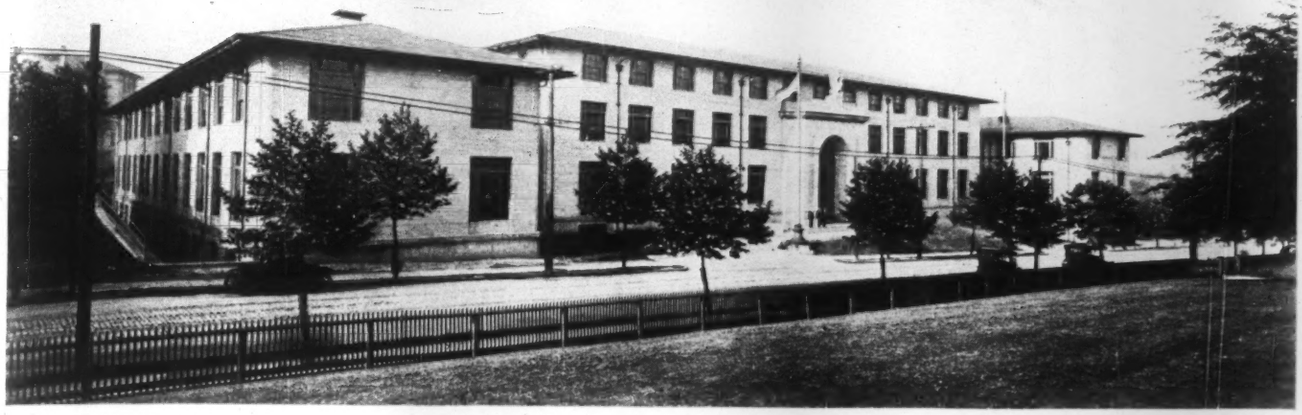
• Heat Transfer Calculations, by Dr. Joseph R. Zwickl. *Refrigerating Engineering*, vol. 34, no. 6, Dec. 1937, pp. 375-382. Mathematical discussion of basic heat transfer formulations.

• Notes on Thermostatic Controls for Coke Fired Boilers, by L. H. Ardley. *Gas Journal*, vol. 220, no. 3885, Nov. 3, 1937, pp. 358-362. Discussion in same vol. 220, no. 3886, Nov. 10, 1937, pp. 537-538. Practical discussion of thermostatic controls with illustration of various control equipment and layout of electric circuits for automatic controls.

COMMITTEE on RESEARCH

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

ANNUAL REPORT—1937



ENTERING its nineteenth year of service to the heating, ventilating and air conditioning industries, the Committee on Research of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS finds itself in a more strategic position than ever before as a coordinator of basic investigations, and as the medium through which vital and unbiased studies of fundamental aspects of these sciences are carried on.

The phenomenal rise of air conditioning to a position of such prominence in the public mind that it is a factor in almost every building operation, enters into every plan for store modernization or industrial plant rehabilitation and is considered by almost every home owner or prospective home owner, has brought with it new opportunities for service, and greater responsibilities.

Today's concepts of air conditioning stem directly from the basic work of the A.S.H.V.E. Laboratory, which for more than a dozen years has constantly been adding to the fundamental knowledge of the art upon which a great industry is being built. In a very real sense therefore, scientific progress in air conditioning and the older arts of heating and ventilating out of which it grew, is synonymous with the growth and development of the research sponsored by this Society.

Out of the work of the past year, there has emerged a clearer realization of the problems which lie ahead and a sharpened conception of the function which the Society can perform in guiding the industry along sound lines. The multiplicity of channels through which air conditioning and related arts is developing and its rapid growth and quickening tempo has substantially increased

[Concluded on page 84]

The 1937 Committee on Research, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

(1) G. L. Tuve, (2) C.-E. A. Winslow, (3) C. E. Lewis, (4) Elliott Harrington, (5) J. H. Walker, (6) F. C. Houghten, Director of Laboratory, (7) H. E. Adams, (8) A. E. Stacey, Jr., (9) W. A. Danielson, Chairman, (10) W. L. Fleisher, Vice-chairman, (11) J. H. Van Alsburg, (12) H. C. Murphy, (13) A. P. Kratz, (14) C. A. Dunham, (15) C. Tasker, (16) D. W. Nelson

Technical Advisory Committee OH-22

COMFORT AIR CONDITIONING

C. TASKER, Chairman

A. E. Beals
F. R. Bichowsky
Thomas Chester
F. E. Giesecke
Elliott Harrington
R. E. Keyes
A. B. Newton
C. P. Yaglou

The work of this committee for the past year included a study of skin temperatures and comfort as they may be affected by a "draft" or air movements of different velocity and temperature, and a study of the relation between a person's feeling of warmth and the effective temperatures maintained in summer air conditioned space. This study shows a definite drop in the skin temperature of the parts affected, varying with the velocity and temperature of the air.

The study of the relation between comfort and the effective temperature maintained in summer air conditioned space was continued in Pittsburgh, Toronto and Texas, somewhat along the lines followed during 1936. A very comprehensive study was made of the reactions of 275 office workers of the Minneapolis-Honeywell Regulator Co., in Minneapolis, to the air conditions maintained throughout the past summer. The small difference between the results found in this study and those previously reported indicates the need for continuing the study in a number of other geographical locations, in order to establish the effects which such variations may have on the conditions under which people may be most comfortable. A paper, "Summer Cooling Requirements of 275 Workers in an Air Conditioned Office," resulting from the Minneapolis study, will be presented at the Annual Meeting.

Technical Advisory Committee OH-3

RELATION OF BODY CHANGES TO AIR CHANGES

E. VERNON HILL, Chairman

N. D. Adams
J. J. Aeberly
John Howatt
A. P. Kratz
P. J. Marschall
V. L. Sherman

A comprehensive program for a study of the application of air conditioning in the treatment of certain diseases was developed about two years ago and a study initiated by Dr. Robert Keeton of the College of Medicine of the University of Illinois, at Chicago. Observations have been made throughout the year, and a paper (Physiologic Response of Man to Environmental Temperature, by Dr. F. K. Hick, Dr. R. W. Keeton and Nathaniel Glickman) is being presented before the Society at the Annual Meeting in New York.

Technical Advisory Committee C-24

CLIMATE AND AIR CONDITIONING

C. A. MILLS, Chairman

G. C. Dunham
James Govan
W. J. McConnell
F. M. Pottenger, Jr.
E. L. Weber
C.-E. A. Winslow
C. F. Neergaard

This committee has been interested in developing a comprehensive program for a study of the effect on persons of various climatic changes in the atmosphere.

It is also continuing the collection of basic data relating to physiological response to physical environment including investigation of the use of radiant heat for comfort air conditioning —both winter and summer. The physiological effects of storm fluctuations in barometric pressure are also being studied.

Few technical advisory committees organized under the Committee on Research, are operating on as broad a base as the Committee on Climate and Air Conditioning.

Set-up for testing physiological effect of drafts in A.S.H.V.E. Laboratory, Pittsburgh

Technical Advisory Committee C-17

TREATMENT OF AIR WITH ELECTRICITY

C.-E. A. WINSLOW, Chairman

R. D. Bennett
W. H. Carrier
L. W. Chubb
W. D. Fleming
R. F. James
L. R. Koller
C. A. Mills
E. B. Phelps
G. R. Wait
W. T. Wells

This is an important committee which is at present reviewing research activity dealing with the application of electricity to the treatment of air for human comfort and health and encouraging work by its members and others.

At present, the most promising approach to this problem appears to be along the lines of changes in the atmosphere due to irradiation which seem to be associated with the stimulation of metabolic activity.

It is the opinion of the committee that the destruction of bacteria in the air by direct application of ultra-violet light and the exclusion of bacteria by a vertical curtain of irradiation have passed the experimental stage and give real promise in hospitals (operating rooms, communicable disease wards, wards for infants) and in certain food industries.

The committee feels that Professor Phelps' studies of the possible effects of irradiation upon bacterial dissociation and mutation are of great interest and hopes it may receive a report from him on this matter.

The committee feels that electrostatic precipitation as a means of removing suspended matter (dust particles and microbes) from air admitted to an occupied space presents great promise.

Technical Advisory Committee IF-31

WEATHER DESIGN CONDITIONS

W. E. STARK, Chairman

E. W. Goodwin
A. C. Grant
J. H. Kincer
A. P. Kratz
L. S. Ourusoff

This committee was recently appointed to outline plans for analysis of Weather Bureau data, with a view of establishing a better understanding of the design outside temperatures for both winter and summer air conditioning. This committee is also interested in a better understanding of the "degree-day" application to heating and cooling problems.

This research was initiated partially in response to a need expressed by the Air Conditioning Code Committee of the A. S. H. V. E., working jointly with the A. S. R. E.

This committee represents an organized effort to make an intensive study of natural weather conditions as affecting design and may eventually influence the form or substance of data now given out by the United States Weather Bureau.

Technical Advisory Committee

HOSPITAL AIR CONDITIONING

T. L. HAZLETT, Chairman

C. J. Barone
B. Z. Cashman
M. B. Ferderber
G. S. McEllroy
H. W. Redding
A. W. Sherrill
C. M. Humphreys

Through cooperation between the Research Laboratory and the Department of Medicine, University of Pittsburgh, a program has been developed for a study of factors entering into the air conditioning of an operating room and a recovery ward. In this connection, an experimental air conditioning system has been provided by the interested industry and installed in connection with an operating room and recovery ward in the Magee Hospital, one of a group of hospitals connected with the University of Pittsburgh Medical School. The installation is near completion, and it is hoped that the University of Pittsburgh, in cooperation with the Laboratory, will be able to collect data throughout the coming year.

The program includes a study of:

(1) Optimum temperature and humidity conditions within the operating room and recovery ward.

(2) The bacteria content of the atmosphere in an operating room as operated in the past, and as operated with the air conditioning system.

(3) The relation between past operation without air conditioning and future operation with air conditioning and controlled ventilation, as it affects the concentration of anesthetic at different points with respect to the operating table.

Technical Advisory Committee IP-4

AIR CLEANING

H. C. MURPHY, Chairman

M. I. Dorfan
C. E. Lewis
S. R. Lewis
G. W. Penne
A. L. Simison
W. O. Vedder

The work of this committee throughout the year has consisted in keeping in contact with and, wherever possible, correlating the efforts of different groups interested in developments in the field of air cleaning. Notable work by Dr. C. B. Coulter and Dr. F. M. Stone, of the College of Physicians and Surgeons, Columbia University, as reported in a recent issue of the *American Journal of Public Health,* indicates that an average of sixteen times as many bacteria are found in unventilated public telephone booths in New York City as in well ventilated booths in the air conditioned building at Radio City, where filtered air is supplied.

Plans for work of the committee during the coming year include analyses of air-borne bacteria found in different types of buildings and their relation to air conditioning applications in general, and methods of filtering or cleaning the air in particular.

Technical Advisory Committee C-12

TRANSPORTATION AIR CONDITIONING

L. B. MILLER, Chairman

T. R. Crowder
F. B. Rowley
A. E. Stacey, Jr.
L. W. Wallace

This committee, appointed during the year, is organizing with a view to developing a comprehensive study of the application of air conditioning to railroads.

The principal purpose of this committee is to cooperate along a broad front with interests identified with railroad operation in order to assist in establishing correct comfort requirements for railroad car air conditioning.

Air conditioned "fever box" developed by A.S.H.V.E. for treatment of venereal diseases

Technical Advisory Committee OH-5

GENERAL A. C. COMFORT REQUIREMENTS

C. P. YAGLOU, Chairman

C.-E. A. Winslow
J. J. Aeberly
R. R. Sayers

Research in the general field of comfort and health has continued during the year with renewed vigor and with considerable expansion in all directions. The Research Laboratory in Pittsburgh and cooperating institutions continued several important studies on drafts and summer cooling, as reported elsewhere by other committees.

The Harvard School of Public Health, in cooperation with the Research Laboratory, published a paper (Ventilation Requirements, Part II, by C. P. Yaglou and W. N. Witheridge, A.S.H.V.E. JOURNAL, July 1937, p. 447), dealing with disappearance of ventilation odors. The evidence presented indicates that the importance of per capita air space on minimum ventilation requirements for the control of body odors arises from the fact that such odors are not stable, but tend to disappear rapidly of their own accord.

Large rooms, it has been found have an advantage over small ones, as they act like reservoirs, allowing body odor to disappear with a minimum of outdoor air supply. The odor of tobacco smoke, on the other hand, not only remains longer in the air, but its intensity increases considerably as the smoke gets stale.

It would appear from such observations as the committee has made to date, that smoking rooms should be made as small as practical and ventilated rapidly. The broad scope of this committee is such that it can to a certain extent, correlate the work of other committees engaged in various phases of comfort air conditioning, although its primary function has been the determination of proper outside air quantities requisite for good indoor air requirements.

Technical Advisory Committee IP-9

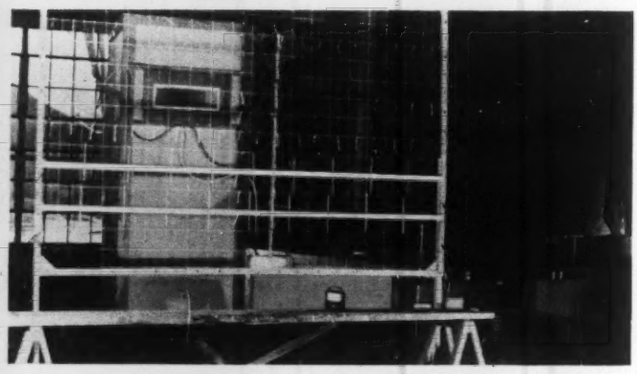
RADIATION WITH GRAVITY AIR CIRCULATION

M. K. FAHNESTOCK, Chairman

B. C. Benson
H. F. Hutzel
J. P. Magos
J. W. McElgin
J. F. McIntire
D. W. Nelson
R. N. Trane
T. A. Novotney

Work under this committee was continued during the year at the University of Illinois in cooperation with the Research Laboratory. A paper (Cooling and Heating Rates of a Room with Different Types of Steam Radiators and Convectors, by A. P. Kratz, M. K. Fahnestock and E. L. Broderick, A.S.H.V.E. JOURNAL, April 1937, p. 251), was presented to the Society at its Semi-Annual Meeting at Swampscott.

Under the program of the committee to be carried on during



Experimental apparatus, Case School of Applied Science, for testing velocity and temperature of entering air within a room

1938-39, the following research should be divided into two parts: (I) That which may be done in the warm wall test booth; and (II) that which may be done in the room heating testing plant.

Under (I) it is hoped that the study of the inlet and outlets of convector heater cabinets with reference to their effect upon the heat output or capacity of the units may be completed in 1938.

Under (II) it is expected that the remodeling of the room heating testing plant will be completed and operations started.

(1) The first tests will be to correlate the performance of conventional tubular steam radiators (sections on 2½ in. centers) in the new 15 ft x 18 ft test room with their performance in the small 9 ft x 11 ft test rooms which were in the old plant.

(2) Tests to be made with conventional 3-tube radiators standing in the room 2½ in. from the plastered surface and placed in recesses below the windows in the exposed walls.

(3) Tests with several different sizes of the new light-weight type of direct steam radiators (sections on 1½ in. centers) to determine their performance, including capacities and the manner in which they distribute heat in the room.

Technical Advisory Committee IP-10

HEAT TRANSFER OF FINNED TUBES WITH FORCED AIR CIRCULATION

F. B. ROWLEY, Chairman

H. F. Hutzel	W. E. Stark
R. H. Norris	G. L. Tuve
C. H. Randolph	C. F. Wood

Work under this committee was carried on at Case School of Applied Science, in cooperation with the Research Laboratory. In a paper (Performance of Fin Tube Units for Air Cooling and Dehumidifying, by G. L. Tuve and C. A. McKeeman, A.S.H.V.E. JOURNAL, June 1937, p. 379) the authors indicated the importance of developing a Code for the testing and rating of dehumidifying coils.

During the year, L. J. Seigel, on a research fellowship, succeeded in working out a very promising analysis which has two important advantages over any other method which has been proposed to date:

(1) Any of the common types of practical problems can be solved directly by formula, without trial and error.

(2) The entire range of performance of the coil can be obtained by running dry-cooling tests only.

A comparative study was made of four methods of computing the performance of a dehumidifying coil, including those published during 1936 by Goodman and by Wells, indicating the advantages of the new method, and steps were taken to check it by a series of tests on one of the coils not previously tested.

Technical Advisory Committee IP-30

COOLING LOAD IN SUMMER AIR CONDITIONING

J. H. WALKER, Chairman

C. M. Ashley	L. S. Morse
John Everetts, Jr.	A. E. Stacey, Jr.
F. H. Faust	R. M. Strikeleather
H. F. Hutzel	

This committee was appointed during the year to study particularly the cooling load in summer cooling and air conditioning for human comfort, resulting from the increased heat transfer through walls and windows due to sun radiation. A comprehensive program was developed with the hope that it could be carried out during the past summer.

This investigation included a study of the heat transfer through 12 sample walls, to be built and studied at the Research Laboratory. These walls were to be subject to sun radiation on the south side and means were to be provided for measuring the heat transfer through the inside surface by a modification of the guarded hot box method.

Technical Advisory Committee IP-21

AIR DISTRIBUTION

ERNEST SZEKELY, Chairman

S. H. Downs	C. H. Randolph
M. K. Fahnestock	J. E. Schoen
F. J. Kurth	G. L. Tuve
D. W. Nelson	J. H. Van Alsburg

This committee, which has been organized and has had plans available for a comprehensive study for several years, found itself for the first time in possession of sufficient funds to make any material progress during the past year. A study was initiated by Professor G. L. Larson of the University of Wisconsin in cooperation with the Society's Laboratory and the Barber-Colman Co., and an investigation of some factors dealing with horizontal distribution of air into an enclosure was made. The study as outlined deals particularly with the relation between the air distribution in the occupied zone to (1) the distance of discharge below ceiling; (2) the aspect ratio of the opening; and (3) the angularity of the blades in grilles. One paper, "Air Distribution from Side Wall Outlets," resulting from the work, has been released for publication and presentation at the Annual Meeting, showing some notable progress.

Technical Advisory Committee IP-20

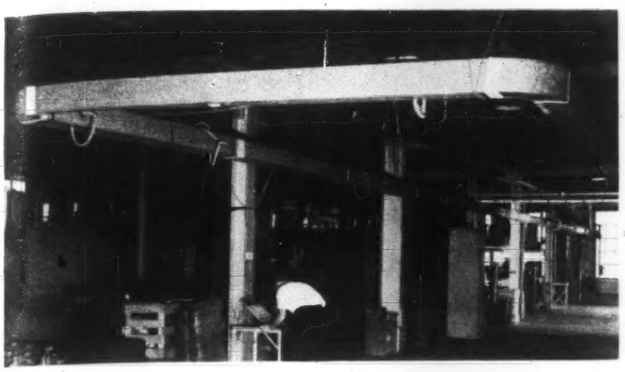
INTERMITTENT HEATING

E. K. CAMPBELL, Chairman

W. L. Cassell	J. M. Robertson
E. F. Dawson	J. H. Kitchen
N. W. Downes	A. H. Sluss
F. E. Giesecke	G. L. Tuve

This committee has been active during the year in analyzing the problem in order to develop a program which may give promise of satisfactory results. Through these efforts, a program involving the cooperation of Professor F. E. Giesecke of the Agricultural & Mechanical College of Texas and Professor E. F. Dawson of the University of Oklahoma has been developed.

This committee is charged in general with a study of the time-rate of heating buildings together with the extra heat load during the heating up period in relation to the structure or type of building.



General view of test set-up in A.S.H.V.E. Laboratory to determine frictional resistance to air flow in small ducts

Technical Advisory Committee IP-7

SUMMER AIR CONDITIONING FOR RESIDENCES

M. K. FAHNESTOCK, Chairman

E. A. Brandt
John Everetts, Jr.
Elliott Harrington
H. F. Hutzel
K. W. Miller
E. B. Newill
F. G. Sedgwick
J. H. Walker
E. D. Milener

Work under this committee was continued at the University of Illinois in cooperation with the Laboratory during the past summer. A paper (Study of Summer Cooling in the Research Residence Using Water at Temperatures of 52 and 46 F, by A. P. Kratz, S. Konzo and E. L. Broderick, A.S.H.V.E. JOURNAL, May and June 1937), resulting from studies carried on during the summer of 1936, was presented to the Society at its Semi-Annual Meeting at Swampscott. A paper (Study of Summer Cooling in the Research Rresidence Using a Small Capacity Mechanical Condensing Unit, by A. P. Kratz et al.), resulting from work carried on during the past summer, will be presented to the Society at the Annual Meeting in New York. The program for the past summer was as follows:

(1) The use of a materially undersized central plant mechanical condensing unit for cooling the first and second floors of the Residence with a unit having only 60 per cent of the capacity that was required to maintain 80 F on the most severe days.

(2) To investigate the problem of cooling the first floor during the day and the second floor during the night, using the small condensing unit.

(3) To obtain additional data on the length of the period between the time that the outdoor temperature reaches a maximum and the time that the heat conducted through the wall reaches a maximum.

Referring to item (2), it is thought desirable that some information on the rate at which the temperature on the second story can be reduced with several different cooling capacities should be obtained. This can be done easily with the condensing unit now installed in the Research Residence by merely changing the speed of the compressor.

The heat-lag and heat-flow studies made under item (3) during the past summer were with an uninsulated frame wall. It is believed that some interesting and useful data can be obtained on an insulated frame wall section by filling the studding spaces with insulation and repeating the test procedure during the summer of 1938.

Technical Advisory Committee IP-6

AIR FRICTION

J. H. VAN ALSBURG, Chairman

C. A. Booth
S. H. Downs
C. M. Humphreys
R. D. Madison
L. B. Miller
L. G. Miller

Plans have been developed for a comprehensive study of the frictional resistance to the flow of air through ducts and fittings, ranging in size from the smallest used in small domestic air conditioning systems to those used in large central fan systems.

A set-up has been developed at the Laboratory in Pittsburgh and a number of preliminary observations made on an 8 in. round, an 8 in. square, and an 8 in. rectangular duct of a 4 to 1 aspect ratio. Considerable delay has resulted from failure to check the results of earlier investigators and the application of the Reynolds Number to the findings. However, these diffiulties have been fairly well accounted for.

In the preliminary work considerable discrepancy or variations were found in the frictional resistance to the flow of air in factory-fabricated duct. These variations have lately been accounted for by a variable friction resulting from constrictions at the joints, which have been found to vary from the equivalent of from 2 to 6 ft of length of duct. It has also been found that the frictional resistance for 8 in. duct is measurably lower than that in use.

Plans for the coming year include a study of factory fabricated and shop-made duct of several different sizes, ranging from 4 to 24 in.

Technical Advisory Committee IF-8

HEAT REQUIREMENTS OF BUILDINGS

O. W. ARMSPACH, Chairman

P. D. Close
W. H. Driscoll
H. M. Hart
V. W. Hunter
H. H. Mather
E. C. Rack
F. B. Rowley
R. J. J. Tennant
J. H. Walker

There has been little change in the comprehensive program developed by the committee over two years ago. The work necessary to follow out these plans is extensive and has engaged all of the research facilities both at Pittsburgh and elsewhere. During the year the Laboratory published another paper (Heating Requirements of an Office Building as Influenced by the Stack Effect, by F. C. Houghten and Carl Gutberlet, A.S.H.V.E. JOURNAL, July 1937, p. 451), completing publication of the results of its study in the Grant Building in Pittsburgh.

Work is being continued at the University of Minnesota on the conductivity of concrete of various types. Professor Giesecke is continuing his studies of the heat requirements of some of the buildings at the Agricultural & Mechanical College of Texas, in cooperation with the Laboratory.

Among the additional activities of this committee is the analyzing of data collected at an earlier date relating to the heating requirements of Sibley Dome, one of the Cornell University Buildings. Other work of the committee has been a study of the degree day method of estimating fuel consumption and a study of condensation and air return rates in the heating of a large school building at the University of Wisconsin.

While this committee has been quite active over a period of years, its current work has been lessened to a considerable degree by lack of finances with which to undertake specific projects. In the matter of cooling requirements for buildings, this phase of research was partially taken over during the year by the Committee on Cooling Load in Summer Air Conditioning, a brief description of whose work will be found in another section of this report. This involves a further study of heat transfer through roofs and walls of buildings as affected by heat capacity as well as solar radiation.

Technical Advisory Committee IF-18

AIR CONDITIONING REQUIREMENTS OF GLASS

M. L. CARR, Chairman

F. L. Bishop
A. N. Finn
E. H. Hobbie
R. J. Lillibridge
R. A. Miller
F. W. Parkinson
W. C. Randall
L. T. Sherwood
J. T. Staples
C. Tasker
G. B. Watkins
F. C. Weinert

This committee, first appointed during 1936, has been particularly aggressive during the year. A sizable budget for the latter part of 1937 and throughout 1938 has been provided for, and a program including the following has been adopted for 1937-1938:

(1) Analysis, distribution to the committee and publication of available current data.
(2) Preparation and distribution to the members of the committee of a comprehensive bibliography.
(3) Cooperation with manufacturers and others carrying on research.
(4) Preparation of monthly reports of progress of the work carried on under the committee's budget for distribution to the committee.

Considerable progress under items (1) and (2) has already been made. At its last meeting a report of a study covering the requirements to heat two small houses, one with single and the other with double glazed windows, was discussed by the committee with a view to preparing a technical paper. Like analyses of several other reports will also be made with a view to the publication of papers and it is the intention of the committee to issue reports on projects at monthly intervals throughout the coming year.

Technical Advisory Committee IF-23

INSULATION

L. A. HARDING, Chairman

E. A. Allcut
H. C. Bates
H. C. Dickinson
J. D. Edwards
E. C. Lloyd
W. E. McMullen
R. T. Miller
E. R. Queer
T. S. Rogers
F. B. Rowley
W. S. Steele
C. Tasker
B. Townshend
G. B. Wilkes

A comprehensive program of research activity was developed by this committee a year ago. This program included a review and a possible revision of the Society's Code for testing insulating materials and heat transmission through walls. During the past year the committee has been engaged in these Code revisions.

Because of the controversial nature of the subject, the work has proved more time-consuming than was anticipated.

Some score of subjects dealing with fundamental aspects of insulation constitute the general work of the committee. These include:

The effect of "regain moisture" on the conductivity of insulating materials.

The effect of condensed moisture on the conductivity of insulating materials in a structure.

Effect of the thickness of an insulating material accepted for test on the conductivity value obtained.

Effect of condensation on the surface of reflective insulation.

Effect of fibre size and fibre arrangement within an insulating board on its conductivity.

Relation between the heat capacity of an insulating material on the rate of heat penetration through a structure in which it is installed.

The degree of settling found for different "fill" materials after it is placed in the structure.

A.S.H.V.E. Research Laboratory Staff
Front Row **(left to right)—F. C. Houghten, Director; David Shore, Research Engineer; Carl Gutberlet, Research Assistant; M. L. Houghten, Technical Librarian; Adolph E. Kurtz, Secretary; Alan C. Byers, Research Engineer**
Second row **(left to right)—Russell G. Hay, Student Assistant; Victor M. Saudek, Student Assistant; Edward Witkowski, Student Assistant; Nicholas Ivanovic, Student Assistant**

Technical Advisory Committee IF-1

SOUND CONTROL

J. S. PARKINSON, Chairman

C. M. Ashley
G. F. Drake
V. O. Knudsen
R. F. Norris
J. P. Reis
A. E. Stacey, Jr.
G. T. Stanton
F. R. Watson
C. H. Randolph

This committee has been active during the past few years in making available the results of research carried on by various members of the committee or others in the industry, and in correlating these results so as to form a sound basis for future research. A comprehensive program for future work has been drawn up, to be carried on as soon as the necessary budget provision has been taken care of.

A paper (The Nature of Noise in Ventilating Systems and Methods for Its Elimination, by J. S. Parkinson, A.S.H.V.E. JOURNAL, March 1937, p. 183), in the nature of a progress report, was presented to the Society at its St. Louis meeting.

Technical Advisory Committee IF-2

CORROSION IN STEAM SYSTEMS

A. R. MUMFORD, Chairman

W. H. Driscoll
C. A. Dunham
L. B. Miller
F. N. Speller
C. M. Sterne
R. R. Seeber

The work of Professor Seeber at Michigan College of Mining and Technology, in cooperation with the Laboratory, was continued during 1937, under a program developed by the committee at an earlier date. Lack of funds somewhat reduced the activity during the latter part of the year.

The following program has been developed for continuation of the work during 1938:

(1) Analyses of condensate produced from steam of known composition in a tight heating system.
(2) Analyses of condensate produced from steam of known composition in an otherwise tight heating system fitted with commercial accessories.
(3) Extent and location of corrosion by oxygen-free condensate containing dissolved CO_2.
(4) Study of patented compounds claimed to be capable of stopping corrosion.
(5) Maintain a case history file of corrosion problems and their remedies.
(6) Cooperation with the *N.D.H.A.* in the revision of standard methods of sampling and analyzing steam and condensate.

Technical Advisory Committee C-11

PSYCHROMETRY

F. R. BICHOWSKY, Chairman

C. A. Bulkeley
J. A. Goff
E. Vernon Hill
F. G. Keyes
A. P. Kratz
W. M. Sawdon

The committee has developed a program for first checking and then, if necessary, recalculating the psychrometric relationships going to make up the psychrometric chart.

It is apparent that certain theoretical errors exist in the data now available; whether or not these theoretical errors are of real practical importance can only be determined by test calculations for a few widely distributed points on the psychrometric chart.

Technical Advisory Committee IP-26

ATMOSPHERIC IMPURITIES AND RESULTING SAFETY AND HEALTH REQUIREMENTS

THEODORE HATCH, Chairman

J. J. Bloomfield
C. A. Booth
Philip Drinker
Leonard Greenburg
Elliott Harrington
H. B. Meller

This committee has been active during the year correlating the work of various investigators interested in the subject.

The general objectives of the committee include

(1) Collection and evaluation of existing published data pertaining to the industrial dust control problem.

(2) Outlining and guiding fundamental research problems by the Society.

(3) Fostering field studies for the collection of basic engineering data.

(4) Action to bring about the preparation and publication of more technical papers in this field. A great deal of valuable engineering work is being done in industrial establishments, but very little of this is described in technical journals in a form that is helpful to others.

Technical Advisory Committee IF-29

EFFECT OF WATER ON ROOFS

A. B. SNAVELY, Chairman

M. R. Beasley
J. B. Griffiths
Elliott Harrington
E. H. Hyde
W. L. Murray
E. R. Queer
C. S. Reeve
E. T. Selig, Jr.

This committee organized during this year and has been interested in a study of the insulating effect of water on roofs, either in the form of pools or in the form of a spray, on the heat transfer outward during the winter, or inward during the summer.

While water sprays on roofs have been fairly common, especially in the West and South, stationary water pools are a relatively newer development. Inasmuch as very little reliable information exists on the results that can be expected from water as an insulation medium on flat roofs, the committee was formed to study all phases of the subject. It is proposed among other things, to secure data in the way of accumulated temperature readings with dry roofs and later with water under similar circumstances.

Cooperating in Research

The following schools, colleges and other organizations cooperated during the year with the A.S.H.V.E. Committee on research, and are continuing their investigations of the various problems listed:

Texas College of Agriculture & Mechanical Engineering—Hot Water Heating, Summer Comfort Cooling Requirements; **University of Wisconsin**—Aeration of Buildings, Effect of Entering Air on Temperature, Velocity and Distribution of Air in Enclosed Spaces; **University of Minnesota**—Air Cleaning Devices, Conductivity of Insulation and Building Materials, Heat Transfer Through Building Construction; **University of Illinois** (College of Engineering)—Direct and Indirect Radiation with Gravity Air Circulation, Residence Air Conditioning, (Medical School)—Air Conditioning in Treatment of Disease.

Case School of Applied Science—Heat Transfer of Finned Tubes with Forced Air Circulation; **Michigan College of Mining and Technology**—Corrosion in Heating Systems; **Ontario Research Foundation**—Summer Comfort Cooling Requirements; **Princeton University**—Measurement of Air Flow Through Ducts; **University of Pittsburgh**—Air Conditioning Requirements for Hospitals; **Yale University** (School of Public Health)—Treatment of Air with Electricity; **Pittsburgh Testing Laboratories**—Air Conditioning Requirements of Glass; **Minneapolis-Honeywell Regulator Company**—Summer Comfort Cooling Requirements.

Contributors to Research

Acknowledgment is made of financial contributions in support of the research work of the Committee on Research, from the following:

General Research

American Air Filter Co., Inc.; American Blower Corp.; Bayley Blower Co.; Barnes & Jones; Bell & Gossett Co.; Crane Co.; Clarage Fan Co.; C. A. Dunham Co.; Detroit Stoker Co.; Fairbanks, Morse & Co.; Julien P. Friez & Sons; Fulton Sylphon Co.; G. & O. Mfg. Co.; Gilbert & Barker; Grinnell Co.; *Heating, Piping and Air Conditioning Contractors National Association;* Hoffman Specialty Co.; Ilg Electric Ventilating Co.; Detroit Lubricator Co.; Young Radiator Co.; Independent Air Filter Co.; Illinois Testing Laboratories; Kewanee Boiler Co.; Kinetic Chemicals, Inc.; Mueller Brass Co.; May Oil Burner Corp.; Modine Mfg. Co.; Minneapolis-Honeywell Regulator Co.; Trane Co.; Timken Silent Automatic; Warren Webster Co.; York Ice Machinery Corp.; Independent Register Co.; J. P. Marsh Corp.; Economy Pump & Machine Co.; Electrol, Inc.; Auditorium Conditioning Corp.; Anemostat Corp. of America; Nash Engineering Co.

Air Conditioning Requirements of Glass

Owens-Illinois Glass Co.; American Window Glass Co.; Mississippi Glass Co.; Franklin Glass Corp.; Pittsburgh Plate Glass Co.; Blue Ridge Glass Corp.; Libby-Owens Ford; The U. S. W. Co.; Unique Balance Co., Inc.; *Metal Window Institute;* Pennsylvania Wire Glass Co.

Residence Air Conditioning

Nash-Kelvinator Corp.

The following firms donated or loaned equipment to the Laboratory and for cooperative research projects:

American Instrument Co., Washington, D. C.; Buffalo Forge Co.; Dravo-Doyle Co., Pittsburgh, Pa.; Julien P. Friez & Sons, Inc., Baltimore, Md.; Excelsior Steel Furnace Co., Chicago, Ill.; Illinois Testing Laboratories, Inc., Chicago, Ill.; Kinetic Chemicals, Inc., Wilmington, Del.; Meyer Furnace Co., Peoria, Ill.; Milcor Steel Co., Canton, Ohio; Minneapolis-Honeywell Regulator Co.; American Air Filter Co., Inc., Louisville, Ky.; Anemostat Corporation of America, New York City; Detroit Lubricator Co., Detroit, Mich.; Acme Industries, Inc., Jackson, Mich.; Westinghouse Electric & Mfg. Co., Pittsburgh, Pa.

[Concluded from page 77]

the complexity of the research problems seeking solution, while at the same time bringing many new subjects forward for investigation.

In these pages are presented brief summaries of the activity of twenty-one technical committees cooperating with the Committee on Research, each investigating some fundamental question of heating, ventilating or air conditioning upon which either new knowledge is required or the improvement of existing data and its revision in the light of new conditions, has become necessary.

That aspect of the problem as a whole which concerns itself with the fundamental human reaction to, and the effect of air conditioning on, human beings, has again been a chief concern of the Committee on Research. In particular, it is meeting a well-defined need for basic information as to the comfort requirements in summer cooling, in order that commercial engineering developments may be directed with more exact knowledge of conditions to be met in practice.

As a natural outgrowth of the continuous study of the physiological reactions of normal people to air conditioning, problems in the application of air conditioning to the treatment of disease are rapidly coming to the fore.

Investigation and study of the therapeutic value of air conditioning has received greater emphasis and it is gratifying to note that the medical profession is becoming more broadly interested in air conditioning than at any time. Evidence of a better understanding and an attitude of mutual respect and cooperation between the physician and the engineer has been especially marked during the year.

A third important aspect of research has been further determination of the effect of air conditioning on the performance of building materials and the extension of this research to the development of new data with respect to various forms of glass insulating and glass building materials which are coming into wider use.

In the division dealing with performance standards, the activity of the year has been concerned with testing the frictional resistance to the flow of air in small ducts, the heat transfer performance of finned-tubes with forced air circulation, direct and indirect methods of radiation with gravity air circulation, the performance of equipment for residence air conditioning, efficiency of air cleaning devices, and corrosion in heating and air conditioning systems, to mention but a few of the subjects investigated.

Looked at as a continuous and progressive program, the year 1937 was marked by the collection of important data which were the means of laying the groundwork for more extensive research during the coming year. It is of interest to note that upwards of 250 individuals, both within and without the membership of the Society, were engaged during the year in the work of the various technical committees and that a number of technical papers were prepared bearing on work initiated by the Committee or carried out in cooperation with engineering schools, colleges and other technical bodies.

Distinguished members of their professions and experts in numerous lines of technical activity have given unstintingly of their time to maintain A.S.H.V.E. Research on the high plane which has characterized it for nearly two decades.

There is gratifying evidence of a broader interest on the part of the industry in general to the importance of A.S.H.V.E. Research, which may be attributed in no small measure to the educational publicity activities carried on during the year.

Respectfully submitted,

W. A. DANIELSON, Chairman,
Committee on Research.

F. C. HOUGHTEN, Director,
Research Laboratory.

BETTER HEALTH OBJECT OF AIR CONDITIONING
Research Engineers Turn From Comfort To New Field
ASHVE Studies Health Phases Of Air Control
Speakers Stress Need For Further Study Of Fundamentals
Air Conditioning Effects Are Studied In Various Fields
"Water Roof" May Become Standard on Future Homes
ENGINEERS SEEKING LEVEL OF COMFORT
'Effective Temperature' Aim of Research Now Being Pushed.
ENGINEERS MAKE STUDY OF DRAFTS
Air Conditioning Survey Started to Secure Data on Air Currents.
Foresee Air Cleaning by Electrical Method
Air-Conditioned Surgery Tested by Medical Leaders
Ventilating Experts Seek Way to Ban Odors From Rooms
Swampscott, Mass. — Engineers may discuss ventilating and air conditioning and use their layman-baffling "cfm's" to describe the flow of air, but what most people want to know is what to do about odors when they think of ventilating.
New Fields Are Seen In Air Conditioning

Educational publicity was carried on during 1937 to familiarize both the industry and the general public with the broad significance of A.S.H.V.E. research. A few sample articles from hundreds of newspapers and magazines mentioning the A.S.H.V.E. Committee on Research are shown in the clippings above

W. E. Heibel, *President*
New York Chapter

Russell Donnelly
Banquet Committee

H. W. Fiedler
Ladies Committee

C. H. B. Hotchkiss
Publicity Committee

Committee on Arrangements

A. C. Buensod
General Chairman

Blank & Stoller
Alfred Engle
Vice-Chairman

44th Annual Meeting A. S. H. V. E.

G. E. Olsen
Entertainment Committee

A. J. Offner
Inspection and Transportation Committee

© Bachrach
W. W. Timmis
Reception Committee

The Big Meeting and Exposition

THE 44th Annual Meeting of the Society will be held at the Hotel Biltmore, January 24-28. Simultaneously the Fifth International Heating and Ventilating Exposition at Grand Central Palace can be visited by members and their friends. During the same week the *American Society of Refrigerating Engineers* will hold its 33rd Annual Meeting and the *National Warm Air Heating and Air Conditioning Association* will have its Mid-winter Meeting at Hotel Roosevelt.

The Society has scheduled six technical sessions for the discussion of important papers and reports, and a feature will be a

Method of Choosing Location of, Financing and Conducting Meetings of the Society

Resolved: That inasmuch as the Annual and Semi-Annual Meetings of the Society come under the jurisdiction of the Council, the following rules governing the handling of such meetings be adopted by the Council and published in the JOURNAL of the Society at least twice during every year, preferably just prior to each meeting:

1—The Council will select the city in which the Annual or Semi-Annual Meeting is to be held, giving due consideration to the invitations received from Chapters or members as well as to the advisability of so distributing those meetings as to make them of the greatest advantage to the general membership, and to reduce as far as possible the expense of members attending.

2—That an appropriation be made to cover the entertainment or local expenses, incurred in connection with the meeting not exceeding $500.00, the regular meeting expense to be taken care of by the General Fund of the Society in the regular way.

3—That no registration fee or compulsory obligations of any nature be imposed on members or guests.

4—That the purchase of tickets for banquets or for any other form of entertainment that may be provided be entirely voluntary.

5—That the grouping of features and the sale of tickets for group features be discouraged.

6—That the raising of funds from manufacturers of heating apparatus be discouraged.

7—That the display of samples, or of literature, advertising the product of any manufacturer in any way, shape or form, be not permitted at the booths, registration desk, or in or about the meetings.

8—That the distribution of trade papers be entirely at the discretion of the committee in charge.

9—That the local Chapter or local members, be empowered to form a General Committee with such sub-committees as may be required to handle the details of transportation, hotel accommodations, entertainment, finance, etc., and that this General Committee be requested to confer frequently with the Council, through the Secretary of the Society, and to make frequent reports on progress in connection with the various matters being handled by them.

10—That the arrangements of elaborate and costly entertainment features be discouraged.

Adopted at Council Meeting, January 29, 1926.

joint session with the *A. S. R. E.* and the *N. W. A. H. & A. C. A.* The Committee on Arrangements has planned some very interesting events for members and ladies. A get-acquainted dinner is planned for Monday evening, January 24, to be held in the Bowman Room of Hotel Biltmore. Advance reservations should be made by all who wish to attend. On Tuesday evening a joint party of the three organizations will be held at the International Casino and a midnight supper will be served. The party starts at 10:00 p. m. and early reservations are advisable. The annual banquet is planned for 7:00 p. m. on Thursday evening in the Biltmore Hotel.

For the ladies, the entertainment program contemplates a shopping tour and style show at 10:30 a. m. on Tuesday at one of New York's leading department stores. On the same afternoon at 2:30 a bridge party at Hotel Biltmore is planned. On Wednesday there will be a tour of Radio City and N. B. C. studios and in the afternoon a theater party will be held. On Thursday the entertainment will consist of a tour of the Italian Liner, Rex, in the morning and in the afternoon the performance at Radio City Music Hall will be enjoyed. The Ladies Committee will supply afternoon and evening passes for broadcasts given by Columbia network. Ladies will be requested to select the events they wish to attend at the time of registration.

The technical program will commence on Tuesday, January 25, when Pres. D. S. Boyden calls the meeting to order. A great variety of subjects will be presented and they will include operating data, research data from the Society's laboratory and other laboratories, and one session will be devoted to physiological requirements in air conditioning. Other sessions will take up such subjects as the heating effect of artificial lighting, ventilating the Lincoln Tunnel, control methods and systems, air distribution, condensation in insulated buildings, cooling towers and applications of steam jet refrigeration, and development of new insulated air duct. The complete program will be found on page 32.

Bowman Room, Hotel Biltmore

The Exposition

At the Exposition in Grand Central Palace the A. S. H. V. E. will maintain information headquarters on the lobby floor and a graphic display of many of its activities in Booths 537 and 538. The Exposition opens at 2:00 p. m. on January 25 and will be open daily thereafter from noon to 10:30 p. m.

Convention Cruise

A convention cruise after the meeting to Bermuda is planned. Details have been furnished to all members. The Furness Line's Monarch of Bermuda will be used for a six-day cruise and special all-expense rates are available to members.

Railroads and air services offer attractive rates to New York and plans should be made now to attend the Society's 44th Annual Meeting and the Fifth International Heating and Ventilating Exposition. Remember the dates—January 24-28!

1. Whitehall Bldg.
2. Downtown Athletic Club.
3. Evening Post Bldg.
4. Telephone Co. Bldg.
5. 88 Greenwich St.
6. One Broadway
7. Harriman Bldg.
8. Adams Bldg.
9. Custom House.
10. Standard Oil Bldg.
11. 50 Broadway.
12. Produce Exchange.
13. Irving Trust Bldg.
14. Continental Bank Bldg.
15. Army Bldg.
16. Woolworth Bldg.
17. Equitable Trust Co.
18. Manhattan Co. Bldg.
19. Empire State Bldg.
20. City Bank-Farmers Trust.
21. 60 Wall Tower.
22. Traders' Exchange Bldg.
23. 99 Wall St.
24. 120 Wall St.

NEWS OF LOCAL CHAPTERS

OFFICERS OF LOCAL CHAPTERS—1938

ATLANTA: *Organized,* 1937. *Headquarters,* Atlanta, Ga. *Meets,* First Tuesday. *President,* E. W. KLEIN, 152 Nassau St., N. W. *Secretary,* C. T. BAKER, 713 Glenn St., S. W.

CINCINNATI: *Organized,* 1932. *Headquarters,* Cincinnati, O. *Meets,* Second Tuesday in Month. *President,* I. B. HELBURN, 610 Chamber of Commerce Bldg. *Secretary,* H. E. SPROULL, 1005 American Bldg.

GOLDEN GATE: *Organized,* 1937. *Headquarters,* San Francisco, Calif. *Meets,* First Tuesday. *President,* B. M. WOODS, Univ. of Calif., Berkeley, Calif. *Secretary,* G. J. CUMMINGS, 113 Tenth St., Oakland, Calif.

ILLINOIS: *Organized* 1906. *Headquarters,* Chicago, Ill. *Meets,* Second Monday. *President,* S. I. ROTTMAYER, 407 S. Dearborn St. *Secretary,* C. E. PRICE, 6 N. Michigan Ave.

IOWA-NEBRASKA: *Organized,* 1937. *Headquarters,* Omaha, Neb. *President,* M. J. STEVENSON, 1643 South 20th St., Lincoln, Neb. *Secretary,* W. R. WHITE, 4339 Larimore Ave., Omaha, Neb.

KANSAS CITY: *Organized,* 1917. *Headquarters,* Kansas City, Mo. *Meets,* Second Monday in Month. *President,* A. H. SLUSS, 827 Mississippi Ave., Lawrence, Kan. *Secretary,* GUSTAV NOTTBERG, 914 Campbell St.

MANITOBA: *Organized,* 1935. *Headquarters,* Winnipeg, Man. *Meets,* Fourth Thursday. *President,* D. F. MICHIE, 492 Wardlaw Ave. *Secretary,* E. J. ARGUE, Ste. 23, Estelle Apts.

MASSACHUSETTS: *Organized,* 1912. *Headquarters,* Boston, Mass. *Meets,* Third Tuesday in Month. *President,* JAMES HOLT, Massachusetts Institute of Technology, Cambridge, Mass. *Secretary,* H. C. MOORE, 69 Massachusetts Ave., Cambridge, Mass.

MICHIGAN: *Organized,* 1916. *Headquarters,* Detroit, Mich. *Meets,* First Monday after the 10th of the Month. *President,* F. J. FEELY, 950 Trombley Rd., Grosse Pointe Pk. *Secretary,* G. H. TUTTLE, 2000 Second Ave.

WESTERN MICHIGAN: *Organized,* 1931. *Headquarters,* Grand Rapids, Mich. *Meets,* Second Monday in Month. *President,* W. W. BRADFIELD, 901 Michigan Trust Bldg. *Secretary,* S. W. TODD, JR., 309 Paris, S. E.

MINNESOTA: *Organized,* 1918. *Headquarters,* Minneapolis, Minn. *Meets,* Second Monday in Month. *President,* R. E. BACKSTROM, Room 1981, First Natl. Bank Bldg., St. Paul, Minn. *Secretary,* F. C. WINTERER, 836 Juno St., St. Paul, Minn.

MONTREAL. *Organized,* 1936. *Headquarters,* Montreal, Que. *Meets,* Third Monday. *President,* G. L. WIGGS, University Tower. *Secretary,* C. W. JOHNSON, 630 Dorchester St., W.

NEW YORK: *Organized,* 1911. *Headquarters,* New York, N. Y. *Meets,* Third Monday in Month. *President,* W. E. HEIBEL, 11 West 42nd St. *Secretary,* T. W. REYNOLDS, 100 Pinecrest Dr., Hastings-on-Hudson, N. Y.

WESTERN NEW YORK: *Organized,* 1919. *Headquarters,* Buffalo, N. Y. *Meets,* Second Monday in Month. *President,* B. C. CANDEE, 19 Tremont Ave., Kenmore, N. Y. *Secretary,* W. R. HEATH, 119 Wingate Ave.

NORTHERN OHIO: *Organized,* 1916. *Headquarters,* Cleveland, O. *Meets,* Second Thursday in Month. *President,* PHILIP COHEN, 401 East Ohio Gas Bldg. *Secretary,* C. A. MCKEEMAN, Case School of Applied Science.

OKLAHOMA: *Organized,* 1935. *Headquarters,* Oklahoma City, Okla. *Meets,* Second Monday. *President,* E. F. DAWSON, University of Oklahoma, Norman, Okla. *Secretary,* E. W. GRAY, Box 1498, Oklahoma City, Okla.

ONTARIO: *Organized,* 1922. *Headquarters,* Toronto, Ont. *Meets,* First Monday in Month. *President,* G. A. PLAYFAIR, 113 Simcoe St. *Secretary,* H. R. ROTH, 57 Bloor St. W.

PACIFIC NORTHWEST: *Organized,* 1928. *Headquarters,* Seattle, Wash. *Meets,* Second Tuesday in Month. *President,* W. W. COX, 326 Columbia St. *Secretary,* M. N. MUSGRAVE, 314-9th Ave., N.

PHILADELPHIA: *Organized,* 1916. *Headquarters,* Philadelphia, Pa. *Meets,* Second Thursday in Month. *President,* L. P. HYNES, 240 Cherry St. *Secretary,* C. B. EASTMAN, 530 Brookview Lane, Brookline, Upper Darby, Pa.

PITTSBURGH: *Organized,* 1919. *Headquarters,* Pittsburgh, Pa. *Meets,* Second Monday in Month. *President,* J. F. COLLINS, JR., 435 Sixth Ave. *Secretary,* T. F. ROCKWELL, Carnegie Inst. Tech.

ST. LOUIS: *Organized,* 1918. *Headquarters,* St. Louis, Mo. *Meets,* First Tuesday in Month. *President,* G. W. F. MYERS, 3947 W. Pine Blvd. *Secretary,* D. J. FAGIN, 1344 Woodruff Ave.

SOUTHERN CALIFORNIA: *Organized,* 1930. *Headquarters,* Los Angeles, Calif. *Meets,* Second Tuesday in Month. *President,* E. H. KENDALL, 1978 S. Los Angeles St. *Secretary,* J. F. PARK, 1234 South Grand.

TEXAS: *Organized,* 1936. *Headquarters,* College Station, Texas. *President,* R. F. TAYLOR, 909 Banker's Mortgage Bldg., Houston, Tex. *Secretary,* W. H. BADGETT, Texas Engrg. Experiment Station, College Station, Tex.

WASHINGTON, D. C.: *Organized,* 1935. *Headquarters,* Washington, D. C. *Meets,* Second Wednesday in Month. *President,* L. OURUSOFF, 411 Tenth St., N. W. *Secretary,* L. F. NORDINE, Room 203, 734 Jackson Pl., N. W.

WISCONSIN: *Organized,* 1922. *Headquarters,* Milwaukee, Wis. *Meets,* Third Monday in Month. *President,* J. H. VOLK, 1906 W. St. Paul Ave. *Secretary,* H. C. FRENTZEL, 3000 W. Montana St.

Atlanta Chapter Appoints Nominating Committee Representatives

December 7, 1937. The Atlanta Chapter, A. S. H. V. E., met in regular session Tuesday evening at the Biltmore Hotel, with Pres. E. W. Klein presiding.

The minutes of the previous meeting were read and approved. Following this Secy. C. T. Baker read a number of communications which his office had received since the last meeting.

President Klein was unanimously elected a delegate to the Annual Meeting of the Society at New York in January, 1938.

President Klein was also unanimously elected as a representative on the Nominating Committee of the Society and Secretary Baker was elected alternate.

Following this there was considerable discussion relative to future technical programs, and it was decided that the delegate to the Annual Meeting would discuss with the national officers the matter of selecting a speaker for a meeting of the Chapter in the early spring.

It was also decided that the date of the January meeting would be changed from the first Tuesday in January to the second Tuesday in order to accommodate some of the members who were expected to be out of the city on the usual meeting date.

W. A. Evans, district manager, Aerofin Corp., upon suggestion of the Secretary, which was concurred in by other members, agreed to deliver a paper at the January meeting, the subject of which will be announced later.

Following this there occurred the usual customary round-table discussion of matters of interest to all.

The meeting adjourned at 9:30 p. m. according to the report of Secretary Baker.

Massachusetts Members Honor Prés. Boyden

December 14, 1937. The regular meeting held at Walker Memorial, Massachusetts Institute of Technology, was preceded by a turkey supper, and was in honor of Col. D. S. Boyden, Boston, president of the A. S. H. V. E.

The principal guest and speaker was President Boyden, who

gave an interesting account of his experiences and visits to other chapters in the United States and Canada. Colonel Boyden then spoke of the Society in general, mentioning its financial condition, membership status, and scope of activities. Later he gave a technical discussion on the Economic Aspects of District Heating, and described some of the installations of the Boston Edison Co. with the aid of slides.

Secy. H. C. Moore reports that the meeting was well attended and many questions were answered by President Boyden at the conclusion of his talk.

November 16, 1937. The November meeting of Massachusetts Chapter was held with a goodly attendance at supper at the Walker Memorial Cafeteria at M.I.T. The business session was held afterward in the Institute buildings, and 48 members were present.

R. G. Stone, of Harvard Blue Hill Observatory and editor of the Bulletin of the *American Meteorological Society*, was the speaker and discussed the subject of Weather. Mr. Stone gave an elementary explanation of physics applied to weather predictions, showing the relation between pressure, wind direction, and temperature as affecting the weather. He then explained the Abercromby type of diagram and later the more recent Norwegian theories. Mr. Stone's talk was illustrated throughout by slides, charts, and weather maps of various types. Secy. H. C. Moore has reported that a discussion followed the address in which several members participated.

Ontario Chapter Discusses Comfort Cooling and Enjoys Pres. Boyden's Visit

December 6, 1937. Eighty-one members and guests attended the Ontario Chapter meeting at the Royal York Hotel, Toronto, according to the report of Secy. H. R. Roth.

Following dinner, W. C. Oke introduced S. L. Elmer, Jr., who spoke on Methods for Comfort Cooling in Ontario. Following this address, J. P. Fitzsimons spoke on the same subject, emphasizing the use of water cooling coils.

An interesting discussion resulted and S. W. Alexander extended a vote of thanks to the speakers on behalf of the Chapter.

An announcement was made by O. L. Maddux that it was necessary for him to resign from the Board of Governors and from the Chapter due to his location in New York.

November 16, 1937. Ontario Chapter held its regular meeting at the Royal York Hotel when dinner was served at 6:30 p. m. and 55 members and guests were present.

Following dinner Pres. G. A. Playfair announced that the Chapter dues, payable May 1, 1938, will be $5.00 and that M. W. Shears will be chairman of the Membership Committee.

Mr. Shears brought to the attention of the meeting that after January 1, 1938 there will be an initiation fee for candidates applying for membership in the A.S.H.V.E.

E. H. Gurney, vice-president of the Society, introduced the principal speaker of the evening, Col. D. S. Boyden, Boston, Mass., president of the A.S.H.V.E. President Boyden spoke on the District Heating System of the Boston Edison Company, and also related briefly the activities and history of the Society. Following President Boyden's address there was considerable discussion on various points which he had brought out.

Secy. H. R. Roth has reported that W. P. Boddington extended on behalf of Ontario Chapter a sincere vote of thanks to President Boyden for his very interesting and instructive talk.

Manitoba Chapter Hears Talks on Use of Gas and Building Construction

November 25, 1937. Pres. D. F. Michie presided at the Manitoba Chapter meeting held at the Fort Garry Hotel, Winnipeg. The minutes of the previous meeting were read and adopted.

President Michie, who had been elected delegate to the Annual Meeting of the Society, withdrew and William Glass was made delegate with R. L. Kent as alternate.

The chairman of the Entertainment Committee, J. B. Steele, reported that plans for an annual banquet on December 11 were formulated.

President Michie read a report on research, and it was suggested that the names of Chapter members interested in this work be forwarded to Col. W. A. Danielson, chairman of the A. S. H. V. E. Committee on Research.

Speaking for the Chapter's Technical Committee, Mr. Steele reported that the Society's Speakers Bureau was arranging to supply a speaker for the January meeting and in February the Winnipeg Sewage Disposal Plant would be discussed by one of the engineers engaged in the construction of this project.

F. Satchwill, gas engineer, Winnipeg Electric Gas Appliance Dept., then gave an interesting address on gas and different types of burners used both in commercial and domestic work.

F. B. Rowley Addresses Manitoba Chapter

October 28, 1937. Pres. D. F. Michie presided at a dinner and meeting of the Manitoba Chapter held at the Fort Garry Hotel, Winnipeg, when Prof. F. B. Rowley, University of Minnesota, Minneapolis, was guest speaker.

The minutes of the previous meeting were read and approved, and upon motion President Michie was appointed delegate to the Society's Annual Meeting in New York in January, 1938. William Glass was then chosen as an alternate delegate.

The President appointed William Worton and Frank Thompson as a committee to make plans to entertain the Minnesota Chapter on the occasion of their visit to Winnipeg in the Spring of 1938.

Greetings from Minnesota were extended by Professor Rowley, and the meeting adjourned to the Kelvin Technical High School where he delivered an address on The Influence of Building Construction on Air Conditioning. Professor Rowley's talk was very interesting to the 80 members and guests who were present, according to the report of Secy. E. J. Argue.

Prof. G. L. Larson Addresses Minnesota Chapter

November 13, 1937. The regular meeting of the Minnesota Chapter was held in the Minnesota Union, University of Minnesota. Dinner was served at 6:30 p. m., after which Pres. R. E. Backstrom called the meeting to order. Approximately 65 members and guests attended.

President Backstrom urged that new members file their applications prior to restoration of initiation fees January 1, 1938.

President Backstrom referred to the 44th Annual Meeting of the Society to be held in New York City during January and encouraged as many members as possible to attend.

A. J. Huch, chairman of the Entertainment Committee, advised that plans were being formulated for a 1937-38 party and that further and ample notice would be given after arrangements had been completed.

It was moved by Prof. F. B. Rowley and seconded by A. J. Huch that President Backstrom represent Minnesota Chapter and serve as Minnesota Chapter member eligible to vote on the Nominating Committee at the 44th Annual Meeting of the Society. The motion was carried.

President Backstrom then turned the meeting over to Professor Rowley to introduce the speaker of the evening, G. L. Larson, professor of Steam and Gas Engineering, and chairman of the Department of Mechanical Engineering, University of Wisconsin. Professor Larson gave an illustrated address on Mechanical Hot Water Heating Systems.

Owing to the increased use and development of mechanical hot water heating systems in the past few years, the subject of Pro-

fessor Larson's topic was of considerable interest and value to Chapter members and guests.

Western Michigan Chapter Enjoys Kalamazoo Meeting

December 13, 1937. The meeting of Western Michigan Chapter was held at the Columbia Hotel, Kalamazoo, and there were 70 members and guests in attendance.

The speaker of the evening was H. E. Birkholz, American Air Filter Co., Louisville, Ky., on the subject of Air Filtration in Theory and Practice. The talk was illustrated with lantern slides. The speaker skillfully pointed out the progress in the field of air filtration, both domestically and industrially. It was his belief that the industrial growth of the country and the inevitable concentration of population had accentuated the smoke and dust problem, and made necessary the elimination of such particles through the use of filters. Mr. Birkholz pointed out the important relationship of contractor to user in the matter of replacement, and stressed the importance of contractor responsibility. The finest particles in the air have yet escaped the operation of present-day filters, but these are too minute to be of much consequence. It was Mr. Birkholz's conclusion that domestic heating and air conditioning would not have made the strides it has today were it not for the fact that the problem of air cleaning has been successfully solved.

Pres. W. W. Bradfield announced the appointment of Clyde Smith and P. O. Wierenga as a committee to consult with the Manager of the City of Grand Rapids in reference to a proposed codification of air conditioning standards for that city. Also, W. G. Schlichting, Kalamazoo, was asked to consult with the proper authority in his city on the same subject, as well as Bruce McLouth of Lansing and Harold Young of Muskegon.

The meeting was well attended, the talk well received, and Kalamazoo proved itself most hospitable. The January meeting will be held in Grand Rapids.

St. Louis Hears Description of Air Conditioning a Windowless Building

October 6, 1937. Pres. G. W. F. Myers called the St. Louis Chapter meeting to order at 7:45 p. m. with an attendance of 23 members and 22 guests.

Secy. D. J. Fagin read an announcement from headquarters office of the organization of the Iowa-Nebraska Chapter, and it was suggested that a letter of welcome be written to this group.

Treas. C. E. Hartwein reported a balance as of October 6, 1937 of $430.63, and President Myers called on standing committees for reports.

E. E. Carlson presented an oral report for the Program Committee and Paul Sodemann told of the meeting held by the Joint Council of the *Associated Engineering Societies* of St. Louis. Mr. Sodemann mentioned the proposed ordinance for the regulation of refrigerating plants and stated that in his opinion the code was of sufficient interest to spend an evening discussing it. Mr. E. A. White, chairman of the Auditing Committee, reported the books to be in order.

President Myers declared a five minute recess and the chair was turned over to Mr. Carlson who introduced R. E. Hattis, consulting engineer, Chicago. Mr. Hattis gave a very interesting and educational talk on Heating and Air Conditioning a Windowless Building. He proved to be an excellent speaker and his description, along with the slides showing hundreds of details involved in conditioning a building of this type, was very well presented. This was evidenced by the number of questions asked following his talk.

President Myers closed the meeting after a rising vote of thanks to Mr. Hattis.

Mechanical Adsorbents Discussed at Illinois Chapter Meeting

November 8, 1937. Seventy-five members and guests were present for dinner and the meeting of Illinois Chapter held at the Brevoort Hotel, Chicago.

Pres. S. I. Rottmayer opened the meeting, and after Secy. C. E. Price had read the minutes of the previous meeting, he appointed A. E. Christopherson and C. M. Burnam, Jr., as tellers for the election of the vice-president.

The following new Chapter members were introduced by E. M. Mittendorff, chairman of the Membership Committee: I. E. Brooke, H. G. Chapin, A. L. Crump, M. J. Fitzgerald, W. B. Schuler, C. A. Storch and R. P. Sunderland.

Tom Brown reported on the activities of the Program Committee and J. J. Aeberly gave the Chapter a brief resumé of the meeting of the Society's Council in Philadelphia.

F. R. Bichowsky, consulting engineer, Dow Chemical Co., was introduced by President Rottmayer and spoke on Mechanical Adsorbents and Their Uses in Air Conditioning. Dr. Bichowsky, a pioneer in the development of liquid drying agents, told the members of early experiments with a number of different chemicals with the view of finding some way of treating air without refrigeration. These various experiments eventually lead to the use of lithium chloride, which according to the speaker, is a good drying agent, harmless to health, not very viscous, and therefore easily pumped around and relatively non-corrosive.

Dr. Bichowsky also presented a chart of the vapor pressure of lithium chloride and showed various types of equipment or contacting surfaces for its use in commercial work. His discussion was informal and interesting and ended with many questions from the audience which served to bring out information on such phases as odors, methods of handling the lithium chloride, etc.

The count of the ballots for vice-president was then announced and resulted in the unanimous election of J. R. Vernon. Mr. Vernon took over his new office and thereby left vacant for nominations from the floor the office of treasurer. For this office, Tom Brown was nominated by J. H. Milliken, seconded by Mr. Aeberly, and inasmuch as Mr. Brown's was the only nomination, it was moved that nominations for the vacancy thus created on the Board of Governors be made at the same time. O. W. Arnspach was the sole nominee for this office. These nominations were filed for ballot at the next meeting.

Golden Gate Chapter Meets at San Francisco

November 9, 1937. A noon meeting of Golden Gate Chapter was held in the Commercial Club, San Francisco. In the absence of Pres. B. M. Woods and Vice-Pres. L. H. Cochran, the meeting was called to order by Secy. G. J. Cummings, and C. E. Bentley acted as secretary pro-tem. It was moved and so ordered that the minutes of the previous meeting be accepted.

Upon motion by Mr. Bentley, seconded by N. H. Peterson, it was voted that the By-Laws as presented by the Committee on Constitution and By-Laws be approved, printed and sent to the members.

G. G. Harrison suggested that some meetings be held on the East side of the Bay in Oakland or at the University of California and that the meetings be held in the evening. A general discussion brought out the fact that about 12 members live on the East side of the Bay and have difficulty in attending noon meetings in San Francisco, especially those at the University. This matter was referred to the Board of Governors for action.

The Secretary was instructed to take up with the Board of Governors the appointment of a Program Committee to have charge of arranging programs and meetings for the Chapter.

W. W. Sandholt, York Ice Machinery Corp., spoke on the development and improvement of modern dichlorodifluoromethane systems with automatic controls. This talk was very interesting both from a historical standpoint and from present day application.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 151 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn, the Council, urge the members to assume their share of responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by January 15, 1938, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
ADAMS, E. E., Sales Engr., Garden City Fan Co., New York, N. Y.	W. T. Wrightson C. S. Pabst	Henry Gitterman W. E. Heibel
ADDINGTON, H. B., Consulting Engr., Brooklyn, N. Y.	R. D. Tyler C. H. B. Hotchkiss	Clifford Strock W. F. Durand (*A.S.M.E.*)
ALLEN, W. A., Sales Engr., Electric Products Corp., Pittsburgh, Pa.	A. J. Zangrilli J. J. La Salvia	R. J. J. Tennant F. B. Mahon
ARMSTRONG, W. J., Consulting Engr., Montreal, Que., Can.	G. L. Wiggs W. U. Hughes	L. H. Laffoley F. J. Friedman
AUCHMOODY, F. W., Air Cond. Engr., Rockefeller Center, Inc., New York, N. Y.	Louis Hament H. L. Alt	C. W. Walton, Jr. H. W. Fiedler
BAILEY, A. E., JR., Sales Engr., H. C. Baker Co., Inc., Roanoke, Va.	A. R. MacMillan F. H. Williams	Albert Buenger Rothwell Woodward
BANOWSKY, A. B., Commercial Sales Director, United Gas Corp., Houston, Texas	C. A. McKinney I. E. Rowe	R. F. Taylor R. W. Kurtz
BECK, DON, Br. Supervisor, York Ice Mach. Corp., Los Angeles, Calif.	H. M. Hendrickson Maron Kennedy	N. W. Berglund Tom Dickinson
BECKER, R. K., Dept. Mgr., Ohio Valley Hdwe. & Roofing Co., Evansville, Ind.	A. C. Willard G. A. Voorhees	L. M. Gram (*A.S.C.E.*) J. D. Thomas (*Non-Member*)
BOWEN, H. C., Treas., Joseph A. Bowen Co., Warren, R. I.	D. S. Boyden E. M. Fenner	R. M. Nee A. A. Beaulieu
BRODNAX, G. H., JR., Htg. Engr., Georgia Power Co., Atlanta, Ga.	E. W. Klein C. L. Templin	L. L. Barnes E. R. Foss
BROWN, J. S., Sales Engr., Frigidaire Corp., Chicago, Ill.	W. R. Knudsen G. W. Lindsay, Jr.	H. J. Zack W. B. Thornton
BURCH, M. J., Htg. Dept., Crane Co., Grand Rapids, Mich.	W. G. Boales S. W. Todd, Jr.	C. H. Morton W. W. Bradfield
BURNAM, C. M., JR., Engrg. Editor, Heating, Piping & Air Cond., Chicago, Ill. (*Advancement*)	J. H. Van Alsburg Herbert Ryerson	R. H. Anderegg William Goodman (*A.S.R.E.*)
CANON, H. A., Mgr. Air Cond., Dravo Corp., Pittsburgh, Pa.	R. J. J. Tennant J. F. Collins, Jr.	T. F. Rockwell C. M. Humphreys
CHAMPLIN, R. C., Asst. to Air Cond. Engr., Timken Silent Automatic Div., Detroit, Mich.	M. C. Kincaide F. R. Bishop	E. B. Root Jay Barton
CHARLES, P. L., Mgr., Walsh & Charles, Winnipeg, Man., Can.	D. F. Michie William Glass	R. L. Kent J. B. Steele
CHERNE, R. E., Engr., Carrier Corp., Syracuse, N. Y. (*Advancement*)	L. L. Lewis W. H. Carrier	J. I. Lyle W. H. Driscoll
CHILDS, L. A., Dist. Sales Mgr., Clarage Fan Co., Philadelphia, Pa.	L. P. Hynes H. H. Mather	W. J. Searle, Jr. R. F. Hunger
CLOSE, ROBERT, Chief Air Cond. Engr., N.B.C., Radio City, New York, N. Y.	C. W. Walton, Jr. A. F. Hinrichsen	C. M. Sterne A. C. Buensod
CLOSNER, J. J., Sales Engr., Robischung-Kiesling, Inc., Houston, Texas	J. A. Kiesling R. W. Kurtz	I. E. Rowe R. F. Taylor
COLMENARES, G. V., Vice Pres. & Mgr., Refrigeracion y Aire Acondicionada, S. A., Havana, Cuba	H. B. Edwards J. F. Des Reis	W. H. Carrier Margaret Ingels
COUGHLIN, R. J., Sales Engr., Gifford Coughlin Co., Royal Oak, Mich.	J. S. Kilner W. G. Boales	F. J. Feely G. D. Winans
COX, P. E., Engr., Societe des Cinetheatres, Hanoi, Tonkin, French Indo China	J. R. Hall L. P. Hanson	C. F. Hawkinson B. P. Edelman
CRANE, R. S., Air Cond. Engr., Delco-Frigidaire Cond. Div., Dayton, Ohio	Albert Buenger C. F. Wood	A. R. MacMillan W. A. Chapman, Jr.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
Cross, R. A., Jr. Engr., Buffalo Forge Co., Buffalo, N. Y.	A. A. Criqui R. D. Madison	C. A. Booth Roswell Farnham
Davis, E. J., Sales Engr., Gurney Foundry Co., Ltd., Toronto, Ont., Can.	Donald Paul E. H. Gurney	J. H. Fox Thomas McDonald
Dimond, R. C., Engr., Airtemp Construction Co., Detroit, Mich.	H. E. Paetz W. H. Old	E. L. Hogan A. P. Darlington
Distelhorst, S. D., Tech. Editor, Heating Journals, Inc., New York, N. Y.	J. D. Hoffman A. W. Williams	Abraham Abrams Clifford Strock
Du Chateau, M. F., Sales Engr., Crane Co., Cincinnati, Ohio.	R. W. Sigmund C. E. Hust	W. C. Pistler H. E. Sproull
Eckstein, J. E., Owner, J. E. Eckstein Co., Pittsburgh, Pa.	J. E. Maynard J. H. Van Alsburg	F. C. McIntosh R. J. J. Tennant
Edwards, L. V., Air Cond. Engr., Cochran Air Cond. Co., Houston, Texas	W. B. Cochran I. E. Rowe	E. P. Renouf I. I. Bock
Ekings, R. M., Jr., Engr., General Elec. Co., Bloomfield, N. J.	D. W. McLenegan Elliott Harrington	F. H. Faust F. O. Urban (*A.I.E.E.*)
Eklund, K. G., Consulting Engr., Karl G. Eklund, Stockholm, Sweden.	C. R. Place H. G. T. Theorell	A. F. Rosell Eric Ohnell (*A.S.S.E.*)
Engdahl, Richard B., Mech. Engr., University of Illinois, Urbana, Ill.	A. P. Kratz W. H. Severns	S. Konzo M. K. Fahnestock
Fedders, M. P., Development Engr., Minneapolis-Honeywell Reg. Co., Minneapolis, Minn.	F. B. Rowley G. D. Kingsland	A. B. Algren L. B. Miller
Foley, J. J., Pres., Weathermakers (Canada), Ltd., Toronto, Ont., Can.	W. C. Oke J. H. Fox	M. W. Shears A. M. Dior
Foster, J. G., Sales Mgr., Air Cond. Utilities Co., New York, N. Y.	H. W. Fiedler Kimball Burr	W. W. Timmis Henry Gitterman
Freitas, L. J., Br. Mgr., Fedders Mfg. Co., Detroit, Mich.	H. F. Hutzel S. A. Whitt	J. A. Lee W. G. Boales
Fuller, E. W., Sales Engr., Delco-Frigidaire Div., General Motors Sales Corp., Dallas, Texas	Herman Worsham Albert Buenger	A. R. MacMillan F. H. Williams
Gabbard, F. W., Sales Engr., York Ice Mach. Corp., Los Angeles, Calif.	H. M. Hendrickson Maron Kennedy	Tom Dickinson N. W. Berglund
Garneau, Leo, Sales Engr., C. A. Dunham Co., Ltd., Montreal, Que., Can. (*Advancement*)	F. A. Hamlet G. L. Wiggs	F. G. Phipps M. W. Shears
Gifford, W. R., Sales Engr., American Radiator Co., Washington, D. C.	W. W. Timmis W. F. Goddard	W. D. Marshall L. S. Ourusoff
Godfrey, J. E., Engr., Delco-Frigidaire Cond. Div., General Motors Sales Corp., Dayton, Ohio	Herman Worsham Albert Buenger	C. E. Lewis C. F. Wood
Goergens, A. G., Mech. Engr., War Dept., Office Quartermaster, Washington, D. C.	M. D. Kiczales Fred Simpson (*Non-Member*)	Joseph Damnhardt (*Non-Member*) C. S. Dean (*Non-Member*)
Green, E. W., Jr. Engr., Green Furnace & Plbg. Co., Lincoln, Nebr.	F. E. Prawl R. F. Rosenbach	M. M. Shapiro C. C. Hellmers, Jr.
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Griffith, J. B., Sales Engr., Drayer & Hanson, Inc., Los Angeles, Calif.	J. F. Park E. H. Kendall	H. B. Lauer A. J. Hess, Jr.
Hamilton, L. L., Office Mgr., Minneapolis-Honeywell Reg. Co., Atlanta, Ga.	R. F. Hahn W. R. Moore	C. L. Templin L. L. Barnes
Harden, J. C., Htg. Engr., Round Oak Co., Dowagiac, Mich.	J. H. Van Alsburg S. H. Downs	T. W. Torr C. R. McConner
Harris, A. M., Sales Mgr., Baker Ice Mach. Co. of Texas, Ft. Worth, Texas.	R. K. Werner W. H. Molar	H. W. Skinner L. L. Landauer
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Heller, J. A., Sales Engr., Air Cond. Utilities Co., New York, N. Y.	H. W. Fiedler Henry Gitterman	W. K. Walker Denis McCormack
Helstrom, C. W., Dept. Mgr., Globe Mach. & Supply Co., Des Moines, Iowa	C. V. Haynes F. H. Gaylord	C. M. Baumgardner E. T. Murphy
Hendrickson, R. L., Consulting Engr., Omaha, Nebr.	W. R. White W. H. Severns	O. A. Leutwiler E. T. Murphy
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Hill, H. G., Sales Engr., Gurney Fdry. Co., Ltd., Toronto, Ont., Can.	Donald Paul E. H. Gurney	J. H. Fox Thomas McDonald
Hinton, R. P., Sales Engr., Crane, Ltd., Winnipeg, Man., Can.	D. F. Michie William Glass	R. L. Kent J. B. Steele
Hoppé, M. F., Mech. Engr., Pierson & Wilson, Washington, D. C.	S. F. Gardner G. D. Fife	A. H. Wilson G. F. Miller
Hopper, J. S., Instructor, A. & M. College of Texas, College Station, Texas	W. E. Long W. H. Badgett	F. E. Giesecke C. W. Crawford (*A.S.M.E.*)
Hotop, N. C., Vice Pres., F. J. Hotop Plbg. & Htg. Co., Kalamazoo, Mich.	W. W. Bradfield C. H. Morton	H. D. Bratt S. W. Todd, Jr.
Huber, Enrique, Mgr., Testamentaria de E. Huber, Ing.-Panuco No. 63, Mexico, D. F.	J. F. McIntire W. A. Russell	R. F. Connell F. H. Blackmore
Hutcheon, C. R., Air Cond. Draftsman, Anemostat Corp. of America, New York, N. Y.	F. J. Kurth T. N. Adlam	E. E. Wells Fritz Honerkamp

CANDIDATES

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JONES, H. L., Zone Engr., Delco-Frigidaire Cond. Div., General Motors Sales Corp., Dallas, Texas

KEELAND, B. W., Sales Engr., Robischung-Kiesling, Inc., Houston, Texas

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KLENERT, WILLIAM, Htg. Supply Sales, Davis & Warshow, New York, N. Y.

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KOCH, A. C., Zone Mgr., Robischung-Kiesling, Inc., Houston, Texas

LADD, DAVID, Br. Mgr., The Powers Regulator Co., Philadelphia, Pa.

LASKARIS, N. G., Student, Ranken Trade School, St. Louis, Mo.

LAWRENCE, L. F., JR., Field Engr., Minneapolis-Honeywell Reg. Co., Atlanta, Ga.

LEWIS, J. C., Sales Engr., York Ice Mach. Corp., Dayton, Ohio

LEWIS, K. C., Air Cond. Engr., Elec. Prod. Corp., Pittsburgh, Pa.

LOCHNER, R. E., Jr. Engr., Delco-Frigidaire Cond. Div., General Motors Sales Corp., Cleveland, Ohio

LOCKWOOD, G. E., Jr. Engr., Howard E. Melton, Inc., Oklahoma City, Okla.

LORE, H. E., Engr., Dravo Corp., Pittsburgh, Pa.

MATHIS, JOHN, Supervisor, Standard Furnace & Supply Co., Omaha, Nebr.

MATHISON, R. S., Asst. Gen. Mgr., Weathermakers (Canada) Ltd., Toronto, Ont., Can.

MCCARTHY, T. F., Div. Mgr., Frigidaire Corp., Chicago, Ill.

MCDONALD, IVAN, Sales, Minneapolis-Honeywell Reg. Co., Winnipeg, Man., Can.

MCGINN, G. F., Surveyor & Installation Supervisor, Baltimore, Md.

MCKEE, J. W., Br. Mgr., U. S. Radiator Corp., Milwaukee, Wis.

MEAGHER, A. T., Director, Wm. Stairs, Son & Morrow Ltd., Halifax, N. S., Can.

MEYER, K. A., Design Engr., L. J. Mueller Furnace Co., Milwaukee, Wis.

MICHIE, D. F., Engrg. Sales Dept., Crane Co., Ltd., Winnipeg, Man., Can. (*Advancement*)

MOREHOUSE, J. S., Prof. Mech. Engrg., Villanova College, Villanova, Pa.

MORGAN, A. S., Mgr., Fess Oil Burners of Can., Ltd., Toronto, Ont., Can.

MORGAN, R. W., Research Engr., Nash-Kelvinator Corp., Detroit, Mich.

MORROW, J. D., Secy.-Treas. & Mgr., The Warren Co., Inc., Houston, Texas

NEWTON, A. B., Control Systems Div. Engr., Minneapolis-Honeywell Reg. Co., Minneapolis, Minn.

NORBY, K. H., Sales Engr., American Radiator Co., Seattle, Wash.

NOYES, R. R., Sales Engr., Canadian Sirocco Co., Ltd., Montreal, Que., Can.

NUTTING, H. G. D., Sales Engr., Vilter Mfg. Co., Detroit, Mich.

O'DOWER, H. J., Sales Engr., Vilter Mfg. Co., Kansas City, Mo.

REFERENCES

Proposers	*Seconders*
Louis Loeffler, Jr. E. W. Gray	J. H. Carnahan Louin Tiller
C. A. Thinn C. M. Brigham	O. J. Prentice C. E. Price
I. E. Rowe J. A. Kiesling	R. W. Kurtz R. F. Taylor
J. A. Kiesling R. W. Kurtz	I. E. Rowe R. F. Taylor
J. L. Forster (*Non-Member*) J. E. Daubney (*Non-Member*)	C. B. Moncrief (*Non-Member*) O. F. Heyden (*Non-Member*)
W. H. Badgett F. E. Giesecke	G. M. Hines W. E. Long
E. J. Richard H. W. Moore	H. E. Sproull G. V. Sutfin
W. P. Culbert R. S. Arnold	C. B. Eastman H. H. Mather
G. E. Olsen F. G. Stellwagen	K. L. Mytinger H. C. Roehm (*Non-Member*)
M. L. Todd M. J Stevenson	F. E. Prawl M. M. Shapiro
I. E. Rowe J. A. Kiesling	R. W. Kurtz R. F. Taylor
L. P. Hynes H. H. Mather	John McElgin A. J. Nesbitt
G. B. Rodenheiser R. J. Tenkonohy	G. W. F. Myers Paul Sodemann
W. R. Moore C. L. Templin	L. L. Barnes W. J. McKinney
H. M. Hendrickson Maron Kennedy	N. W. Berglund Tom Dickinson
J. J. La Salvia A. J. Zangrilli	F. B. Mahon G. E. Lynn (*A.S.R.E.*)
A. R. MacMillan F. H. Williams	L. T. Avery R. A. Wilson
F. H. Williams A. R. MacMillan	E. W. Gray C. Z. Woodworth (*Non-Member*)
R. J. J. Tennant F. C. McIntosh	F. B. Mahon D. W. Loucks
F. R. Bishop W. R. White	R. K. Milward W. E. Nesbit (*Non-Member*)
W. C. Oke G. A. Playfair	M. W. Shears A. M. Dion
G. W. Lindsay, Jr. W. R. Knudsen	H. J. Zack W. B. Thornton
J. H. Fox Thomas McDonald	William Worton D. F. Michie
C. W. B. Fordney (*Non-Member*) R. B. Graeves (*Non-Member*)	J. B. Gaulel (*Non-Member*) Samuel Meisel (*Non-Member*)
J. H. Volk C. H. Randolph	H. C. Frentzel M. E. Erickson
E. H. Gurney H. B. Jenney	William Philip T. H. Worthington
E. A. Jones Ernest Szekely	C. H. Randolph R. G. Koch
William Glass J. B. Steele	E. J. Argue R. L. Kent
L. P. Hynes H. H. Mather	C. B. Eastman L. A. Tucker
Thomas McDonald H. B. Jenney	H. R. Roth J. H. Fox
Herbert Van Nouhuys A. L. Hesselschwerdt	S. A. Whitt H. F. Hutzel
J. A. Kiesling I. E. Rowe	R. F. Taylor C. A. McKinney
F. C. Houghten G. D. Kingsland	L. B. Miller C. W. Nessell
R. D. Morse S. D. Peterson	R. O. Wesley M. N. Musgrave
C. W. Johnson A. W. Peart	F. G. Phipps Leo Garneau
S. S. Sanford G. D. Winans	G. H. Tuttle E. E. Dubry
N. W. Downes A. L. Maillard	F. J. Dean, Jr. Benjamin Natkin

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
OELGOETZ, J. F., Owner, J. F. Oelgoetz Co., Columbus, Ohio.	D. H. Wyatt W. B. Cooper	E. D. White D. B. Poling
OKE, W. C., Air Cond. Engr., Weathermakers (Canada), Ltd., Toronto, Ont., Can. (*Advancement*).	M. W. Shears A. M. Dion	G. H. Hopper J. H. Fox
PARENT, H. M., Partner, Parent & Kirkbride, Philadelphia, Pa.	W. P. Culbert R. S. Arnold	C. B. Eastman H. H. Mather
PARKER, P. E., Htg., Vtg. & Air Cond., H. C. Baker Co., Roanoke, Va.	A. R. MacMillan G. B. Houliston	P. H. Beam (*Non-Member*) W. D. Ehlers (*Non-Member*)
PELLMOUNTER, T. V., Student Engr., Worthington Pump & Mach. Co., Kansas City, Mo.	E. K. Campbell E. K. Campbell, Jr.	Thomas Pellmounter L. A. Stephenson
PESTERFIELD, C. H., Instructor, Mich. State College, East Lansing, Mich. (*Advancement*)	F. B. Rowley L. G. Miller	B. F. McLouth A. B. Algren
PFEIFFER, F. F., Designer, United Engrs. & Constructors, Inc., Philadelphia, Pa.	H. R. James Pierce Timmis	R. F. Hunger L. C. Davidson
PHILLIPS, W. L., Mgr. Airtemp Div., Griffith Consumers Co., Washington, D. C.	H. M. Rudio T. H. Urdahl	F. M. Grimes J. G. Vaughan, Jr.
PLEUTHNER, R. L., Engr., Buffalo Forge Co., Buffalo, N. Y.	W. R. Heath A. A. Criqui	R. D. Madison F. D. Hart
POWELL, G. W., JR., Mech. Engr., George W. Powell, Jr., Philadelphia, Pa.	Pierce Timmis H. R. James	C. B. Eastman R. F. Hunger
PREBENSEN, H. J., Vice Pres., Air Comfort Corp., Chicago, Ill.	E. M. Mittendorff E. P. Heckel	W. F. Slater E. T. Murphy
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WHITE, W. R., Mgr. Air Cond. Dept., Nebraska Power Co., Omaha, Nebr. (*Advancement*).	J. E. Davidson (*Non-Member*) F. E. Smith (*Non-Member*)	Clarence Talsma (*Non-Member*) Fred Downs (*Non-Member*)
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WOODWARD, ROTHWELL, Sales Engr., Delco-Frigidaire Cond. Div., General Motors Sales Corp., Dayton, Ohio	Herman Worsham Albert Buenger	C. E. Lewis A. V. Hutchinson

Candidates Elected

In past issues of the JOURNAL of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Committee on Admission and Advancement and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. B-III, Sec. 8, of the By-Laws, the following list of candidates elected:

MEMBERS

BJERKEN, M. H., Sales Engr., Hoffman Specialty Co., Minneapolis, Minn. (*Advancement*)

BROOKE, I. E., Consulting Engr., Chicago, Ill.

COOLEY, E. C., Mfrs. Repr., San Francisco, Calif.

DIETZ, C. F., Sales Engr., Haynes Selling Co., Philadelphia, Pa.

FEYGE, HAROLD, Mgr. Htg. Dept., Walworth Calif. Co., San Francisco, Calif.

FITZ, J. C., Mgr. Arco Thermo System Div., American Radiator Co., New York, N. Y. (*Reinstatement*)

HENNESSY, W. J., Engr., Green Fdry. & Furnace Wks., Lincoln, Nebr.

HILL, J. A., Air Cond. Engr., Pacific Gas & Elec. Co., San Francisco, Calif.

HOLLAND, R. B., Sales Engr., York Ice Mach. Corp., San Francisco, Calif.

HOOK, F. W., Br. Mgr., Johnson Service Co., San Francisco, Calif.

JOHNSON, O. W., Chem. Engr., Standard Oil Co. of Calif., San Francisco, Calif.

KAUP, E. O., Engr., Air Cond. Div., W. R. Ames Co., San Francisco, Calif.

LANDAUER, L. L., Consulting Engr., Kribs & Landauer, Dallas, Texas (*Advancement*)

MAYNE, W. L., Br. Mgr., U. S. Radiator Corp., Cincinnati, Ohio

MCGAUGHEY, H. M., Sales Mgr., Nash Kelvinator Corp., Detroit, Mich.

MEYERS, JOHN, Br. Mgr., Johnson Service Co., Washington, D. C.

MILLHAM, F. B., Installation Mgr., S. S. Fretz, Jr., Inc., Philadelphia, Pa.

NIESSE, J. H., Ind. Dist. Mgr., Ilg Elec. Vtg. Co., Indianapolis, Ind.

ROSE, H. J., SR., Industrial Fellow, Mellon Inst., Pittsburgh, Pa.

SEARLE, W. J., JR., Air Cond. Engr., The Ballinger Co., Philadelphia, Pa.

SHERBROOKE, W. A., Mgr. Technical Div., Utica Radiator Corp., New York, N. Y.

STANTON, H. W., Comm. Sales Dir., Iowa-Nebraska Light & Power Co., Lincoln, Nebr.

STOTT, F. W., Sales Engr., C. A. Dunham Co., Ltd., Toronto, Ont., Can.

WETHERED, WOODWORTH, Consulting Engr., Hotel Sir Francis Drake, San Francisco, Calif.

WHEELER, JOE, JR., Sales Repr., Johnson Service Co., New York, N. Y.

ASSOCIATES

ALLAN, WILLIAM, Pres. & Treas., Allan Engrg. Co., Milwaukee, Wis.

BELSKY, G. A., Air Cond. Engr., Majestic Refrigerator Corp., New York, N. Y.

CUNLIFFE, J. A., Western Can. Mgr., A. P. Green Co. of Can., Ltd., Winnipeg, Man., Can.

HAUER, FRED, Pres., Fred Hauer & Co., Peoria, Ill.

HIGDON, H. S., Sales & Estimating, Andrews Heater Co., San Francisco, Calif.

HOLYFIELD, E. F., Air Cond. Engr., Okla. Electrical Supply Co., Oklahoma City, Okla.

KNOWLES, F. R., Director Comm. Engrg. Dept., Penn. Elec. Co., Johnstown, Pa.

MIDEKE, J. M., Sales, Mideke Supply Co., Oklahoma City, Okla.

MILES, C. N., Foreman, Kohlenberger Engrg. Corp., Fullerton, Calif.

MURNIN, E. A., JR., Dept. Head, Sarco Mfg. Co., Bethlehem, Pa.

O'CONNOR, G. P., Pac. Coast Div. Mgr., The Ric-wiL Co., San Francisco, Calif.

SIDWELL, E. W., Sales Mgr., Armstrong Furnace Co., Columbus, Ohio

SOLZMAN, I. I., Owner, Pasol Engrg. Co., Omaha, Nebr.

STIEGLER, A. J., Owner, Valley Sheet Metal Works, Neenah, Wis.

WHITE, T. G., Jr., Sales Engr., U. S. Radiator Corp., Louisville, Ky.

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CONSTANCE, J. D., Draftsman, York Ice Mach. Corp., Brooklyn, N. Y.

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FISCHER, LADISLAV, Engr. in Charge of Mfg., Anemostat Corp. of America, New York, N. Y.

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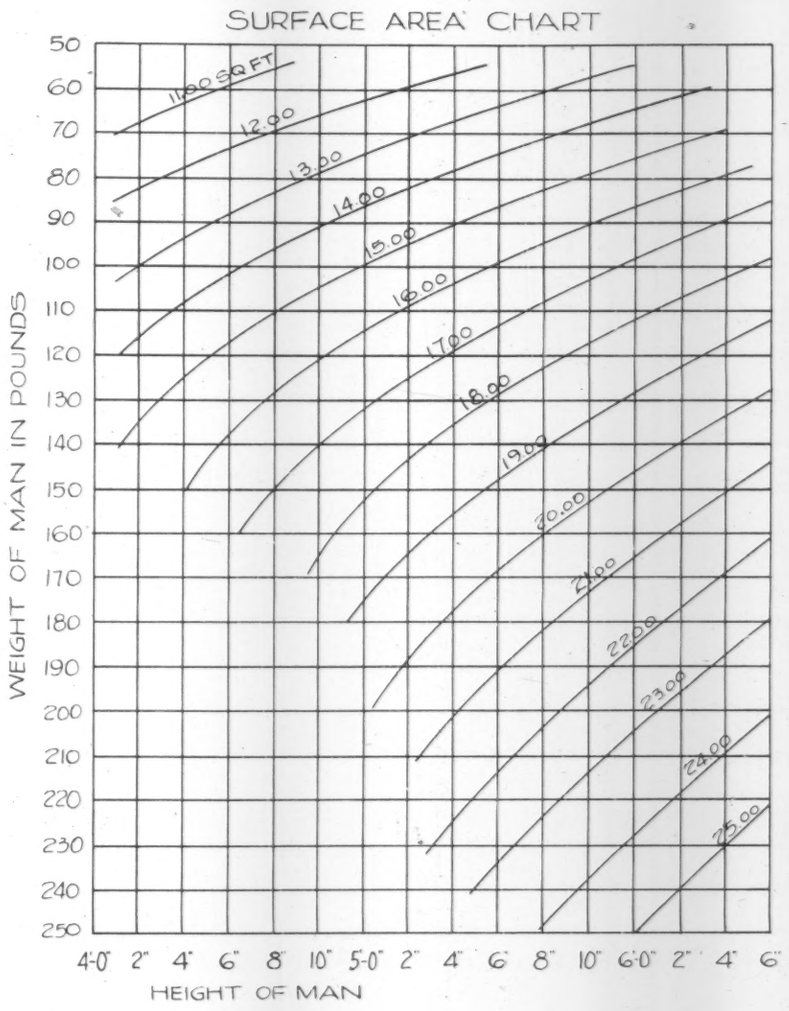
WHITE, T. J., Sales Engr., American Blower Corp., San Francisco, Calif.

WILLIAMS, D. D., Gas Htg. Engr., Iowa-Nebraska Light & Power Co., Lincoln, Nebr.

STUDENTS

LONGWELL, J. C., Student, Mass. Inst. of Tech., Cambridge, Mass.

E. Vernon Hill's* Personal Comments



Body Heat

THE important and interesting problem of human comfort in relation to the air environment has occupied the attention of the A.S.H.V.E. research laboratory more or less continuously since it was founded, and has been investigated by physiologists and research workers for the past 150 years.

In 1776, Lavoisier found that the atmosphere contained two gases, azote (nitrogen) and a highly respirable air, oxygen. It was he who first realized the importance of Priestley's discovery of oxygen only a few years before. Lavoisier not only first appreciated the significance of oxygen in respiration but, with the aid of Seguin, developed a respiratory apparatus which measured with considerable accuracy the metabolism of man. He also demonstrated most of the basic factors regarding the increase of heat production caused by food, by exposure to cold, and exercise.

From this time to the present, work has been more or less continuous on the part of physiologists to explain the important but obscure phenomena of heat production and dissipation by the human body. The results of this research work have been interesting, in many cases surprising, and of course extremely valuable.

The belief of many engineers is that body heat production is changing constantly with changing air temperatures. This, however, does not seem to be the case. The amount of heat generated in the human body may be, for the purposes of explanation, divided into two parts: (1) The heat necessary to maintain body organs and functions in operation, called the *basal metabolism;* and (2) the heat generated by muscular activity.

The basal metabolism is determined by measuring carefully the oxygen in the inspired air with a subject at absolute rest and without food for the preceding 12 to 16 hours. This so-called basal metabolic rate is surprisingly constant in man and other warm blooded animals, winter and summer, day and night and under considerable variations of temperature. It does not vary uniformly according to the weight of a person as one would suspect, but it seems to be determined by the surface area. (To be exact, we should not say that it is "determined" by the surface area, but for some unknown reason follows it very closely). Physiologists believe that there is some fundamental condition that seems to connect them—what it is we do not know.

Lusk gives a table of metabolism for different animals based on heat loss per kilogram of weight and per square meter of body surface. Transposed into English

*Director of Research and Education, Refrigeration and Air Conditioning Institute. Member of Board of Consulting and Contributing Editors.

measurements, it is as follows:

	1 Weight Lb	2 Surface Sq Ft	Btu per Hour		5 Total Btu
			3 Per Lb	4 Per Sq Ft	
Horse	975.0	59.5	0.96	15.75	935
Pig	283.0	27.70	1.63	16.65	462
Man	142.0	23.6	2.68	16.15	382
Dog	33.6	9.0	4.29	16.10	145
Goose	7.75	2.87	5.55	14.95	42.9
Rabbit (without ears)	5.1	2.24	6.25	14.25	31.8
Fowl	4.4	1.66	5.92	15.65	26
Mouse	.40	.38	17.50	18.40	7

It will be noted that if the metabolism is based on the weight (column 3), there is a wide variation in the heat production per pound, but when it is based on the skin surface (column 4) there is a surprising consistency. Various explanations are advanced for this interesting phenomena. It is believed that the mass of the cell structure of the body follows closely the surface area and that all cells being subjected to uniform conditions at all times, give off the same amount of heat. Therefore, under basal conditions, the total heat dissipated varies as the body surface area.

The surface area chart reproduced here is adapted from DuBois, rearranged in English units. It takes into consideration weight as well as height, and by using an empirical formula makes it possible to determine the skin surface with an error of less than 2 per cent.

The basal rating—that is, the minimum amount of heat required to maintain the body temperature and functions—is approximately 16 Btu per sq ft of surface. Moderate activity raises the rate to 30 and hard work to 50. It is thus evident that a little over half of the heat generated in the body is a constant quantity for each individual, depending upon the skin surface area. It does not vary at different times or at different seasons of the year, or even in changing air temperatures. It is not greatly affected by clothing or other ordinary variations in environment. It *is* affected by starvation and by certain diseases, notably those of the thyroid gland.

Heat production *above the normal metabolic rate* is increased after partaking of food and it is materially increased by muscular exertion.

Certain logical (not experimental) conclusions may be formed from this data. It does not appear that comfort is directly influenced by the amount of heat generated in the body. The metabolic rate has no effect on comfort. The heat production from the digestion of food and from muscular exertion only indirectly affects comfort in affecting the skin temperature and the latter is determined far more by the clothing than by any other factor. It seems to follow from the foregoing that the air temperature and humidity has little effect in influencing heat production or dissipation in the body. The greater portion of our feelings of comfort in relation to the air environment is a sensory impression of the nerve endings in the skin not influencing heat production and only to a very minor extent, heat dissipation from the body.

The acceptance of this view point indicates certain conclusions regarding human comfort in relation to the air environment. Comfort is sensory and to a certain extent psychic. It is markedly influenced by the clothing we wear, by the food we are digesting, and by our muscular activity. The effect, within the short range of ordinary comfort temperatures, from the air environment is not as great as has frequently been supposed and *definite* comfort conditions cannot be predicted from the temperature and humidity of the surrounding air.

Schools and Study Courses

Extension Course in Technical Writing Revised

The extension division of the University of Wisconsin, Madison, has announced a thorough revision of its home study course in composition of technical papers.

The course is designed to answer the need of technical men for a proper mastery of the English language in order to write for publication clearly and effectively. It presents in a few assignments the essential principles of English composition, introducing the student to an effective professional style, to the planning and arrangement of technical papers, to the important points in paragraphing, punctuation, capitalization, abbreviation, etc.

Adds Courses in Stokers

The Herkimer Institute, 1819 Broadway, New York, N. Y., has added courses in stokers to its curriculum. Both day and evening instruction is offered; personal attendance of the student at the lecture rooms and shops of the Institute is required.

Among the courses available are refrigeration; air conditioning and refrigeration; oil burners and automatic heating; coal stokers and automatic heating; oil burners, automatic heating and air conditioning; coal stokers, automatic heating and air conditioning; fuel utilization; and heating and ventilating. These subjects are treated so as to be particularly adaptable to the needs of installation and service men.

Book Reviews

Weather Elements

["Weather Elements—A Text in Elementary Meteorology," by Thomas A.Blair, Senior Meteorologist, U. S. Weather Bureau; Assistant Professor of Meteorology, University of Nebraska. 1st ed. 1937 401 + xv pp., 6x9 in., clothbound. Published by Prentice-Hall, 70 Fifth Ave., New York, N. Y. Price, $5.00.]

An important secondary object of the author is to present in this new work "that general body of information about the weather and the present state of our knowledge concerning it, which it is believed that every intelligent person should possess in relation to this most important element of his environment." The work of the heating and air conditioning engineer is so intimately associated with the weather, and so much of his data are based upon the meteorologist's observations, that a book of this kind is particularly pertinent. It comprises an introduction to the science of meteorology in its present state of development.

Chapters are included on the atmosphere; observing temperature, pressure, and wind; observing moisture, sunshine, visibility, and upper air conditions; solar radiation and its effects; condensation of water in the atmosphere; inter-relation of temperature, pressure, and wind; the general circulation; the secondary circulation; lesser disturbances; weather forecasting; world weather; climate; climate and man; electrical and optical phenomena; and the United States Weather Bureau.

Forced Circulation Water for Heating and Cooling

By Samuel R. Lewis*

THIS is a record of some of the features applied during recent years in year around air conditioning by my office. It refers particularly to the Jeffrey Mfg. Co., makers of mining and conveying machinery.

In this large general office installation the entering air is warmed in winter by four rows of finned tubes using warm water applied so that the entering water heats the tubes first exposed to the cold air. The water is circulated mechanically through a controlled steam transfer heater and is treated with an automobile type antifreeze solution as a final precaution against accidental freezing. When warm weather approaches, cold water from a deep well is circulated, this time through eight rows of finned tubes in a counterflow direction, so that the coolest water enters the leeward side of the convectors.

The building is exposed on all four orientations, is four stories high and has about 100 rooms of varying size with widely different heat transfer requirements, both latent and sensible. There are automatically controlled direct radiators and duct reheaters for every room so that very satisfactory temperature control is in effect in either very cold weather or very hot weather. Automatic control of relative humidity is provided the year around. A gas burning auxiliary boiler provides warm water for the duct reheaters in very warm weather to permit shutting down the large steam boilers used in winter.

This plant has been installed in three stages in three successive years—first the third story, then the first and second stories, and last the fourth story. This has given opportunity to correct unforeseen matters as each addition was made. The president of the Jeffrey company states that experience with the plant has proved the heavy investment to have been eminently justified, as evidenced by reduction in absenteeism and improvement in efficiency of all employees.

Operation in Spring and Fall Periods

With the overhead convector units used for the first installation it was difficult to read thermometers and gages, and maintenance of valves and dampers (even though they were nearly all of the mechanically operated type) was awkward and somewhat dangerous. There was an occasional drop of water or oil from bearings or glands which was most objectionable. As a result, space cheerfully was given by the owner when the second installation was made so that the convector units could be placed in rooms devoted to their sole occupancy, in the first story, with complete and easy access to all controls. A penthouse on the roof was built especially to house the third installation, the cost being justified in the mind of the owner, to secure safe drainage and access.

The deep well water has a high proportion of precipitate, so that when the seasonal change from cooling to heating takes place it is necessary to substitute city circulating water in order to prevent a heavy coating of salts on the water heating surfaces. The change in water, with washing out, requires some time, so that when the system is once changed from using cold well water to hot city water it cannot quickly and conveniently be switched back to the use of cold well water.

There are periods of perhaps two months during every spring and fall when heat may be required one day and refrigeration the next. Indeed, we find in this densely occupied four orientation building with heavy use of electric lights that heat may be required early in the morning on some days, with refrigeration needed by noon. The original design, using hot water duct reheaters from an auxiliary gas fired boiler, answered admirably the need for heat and control during mild weather for the rooms not receiving sufficient warmth from the sun and the electric lights, but if the warm weather reappeared after the well water cooling had been discontinued, with city water in the system, the plant failed to furnish refrigeration for the warm zones. These zones of course changed constantly under varying reception of sensible heat from the sun and the lights. This matter was corrected by installing a steam-using convector to windward of the water-using convectors, so that the well water convectors could be kept in readiness to serve at all times until the weather man made up his mind definitely that the heating season had arrived.

Under the present arrangement, if the outside air is too warm to accomplish adequate cooling, any room in this building can be cooled during any intermediate-temperature season at the same time any other room is being heated, all under automatic temperature control and all with reasonable economy. On a frosty morning in October the steam convectors of the six central type convector systems which serve the entire building will warm the air to 70 F and the duct reheaters—a separate one for every air delivery diffuser—will pass it unheated or will warm it as may be needed to any degree between 71 and 85 F under control of the individual thermostats. As the sun gets in its work and as the other sensible heat inflow becomes manifest, an outdoor thermostat readjusts the master supply duct thermostat, which closes off the steam convectors and starts the cold water pump so as to give a general air supply temperature of 60 F properly dehumidified under bypass control. The duct reheaters continue to pass the air unheated or will warm it as may be required without any readjustment of the thermostats. This type of system has been copied in its essential principles for hundreds of rooms in many office buildings and residences with universal satisfaction.

The peculiar complication due to "unheatable" well water is of course unique. The deep well water in

*Consulting Engineer. Member of Board of Consulting and Contributing Editors.

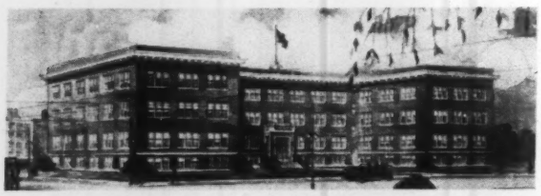
It would be valuable, were it possible, to present the trends in thinking of every engineering office engaged in air conditioning work—as the author does here for his own office. Mr. Lewis is inclined toward the use of forced circulation water for heat conveying, using cold water in summer and warm water in winter and utilizing as much of the equipment as possible for both heating and cooling service. . . . To show how his ideas work out in practice, he describes an installation for the general offices of the Jeffrey Mfg. Co. at Columbus, Ohio, where cold well water is available. The design was complicated by the heavy precipitate content of the water which deposits on any water heating surfaces

Columbus is very cold, but carries a heavy burden of impurities with so much odor that the health department refuses to permit its discharge into the sewer in some parts of the city, since ultimately the water reaches the ponded river and leads to an unjust suspicion of putrefaction there. Therefore, the custom in Columbus for disposal of the water used in cooling systems is to allow it to run back to a second well. When this bad water is heated or even allowed to settle for any appreciable length of time there is a deposit of solids which closes the water circulating passages quickly and which can be dislodged only by chiseling or reaming.

When a mechanical refrigeration system is used in place of well water the temperature control measures described above will still be necessary; that is, there must be ability to make an almost instant switch from cooling to heating, with separate steam or water heating convectors in each convector unit, each having separate additional direct expansion or chilled water surfaces. With duct reheaters as described the same results could be obtained as with the cold well water. We have used in many plants eight rows of finned tubing, with four rows for heating and eight for cooling, all with forced circulation water as the year around heat carrier. We have refrigerated for, say, ten degrees clean well water which was too warm to give adequate dehumidification, and have employed duct reheaters which gave individual room temperature control simultaneously when heating the entire plant or when cooling it.

We have been embarrassed to a greater or less degree on many days during September, October, April, and May, because some rooms needed refrigeration while 100 per cent outside air was not cool enough to do the cooling at the same moment that other rooms needed heating.

The Jeffrey plant described carries one answer. With the lessons of this and other jobs in mind the practice of my office is toward the use of automatic air filters, shuns overhead convector units, and leans toward forced water for heat transfer. We prefer large central air supply systems with duct reheaters rather than to use several separate small zone convector units which usually must be rather inaccessible. We are not particularly happy about systems in which the air supply to some of the rooms must be throttled in order to secure individual temperature control, preferring to employ constant air volume and varying temperature rather than constant temperature and varying air volume.



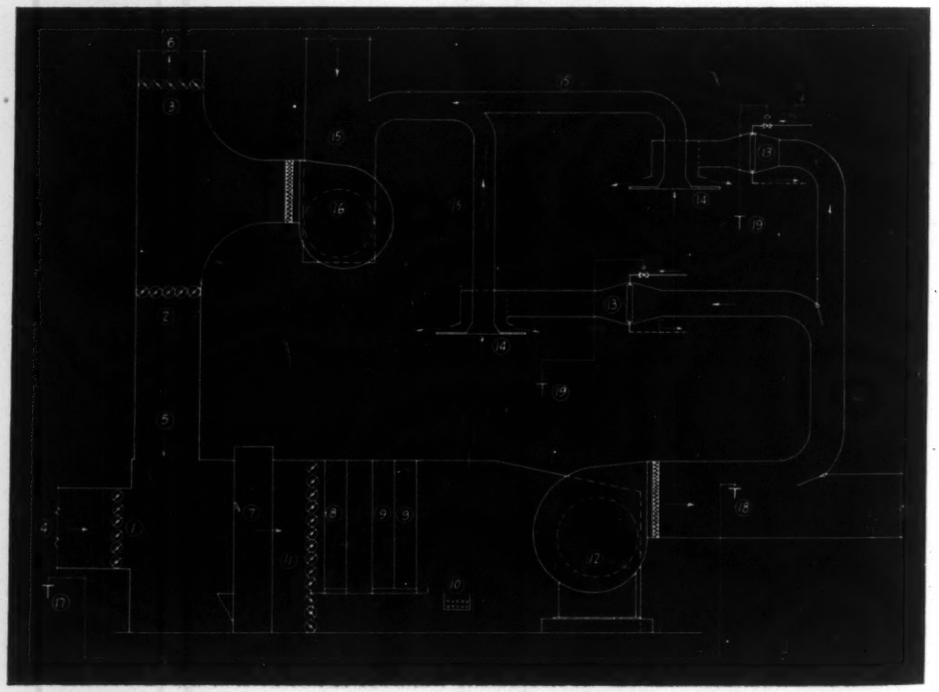
At the left is a view of the general office building of the Jeffrey Mfg. Co., in which year 'round air conditioning was installed "step-by-step." Forced circulation water is used winter and summer. Below is a simplified diagram of the air conditioning system showing the method of air circulation and control

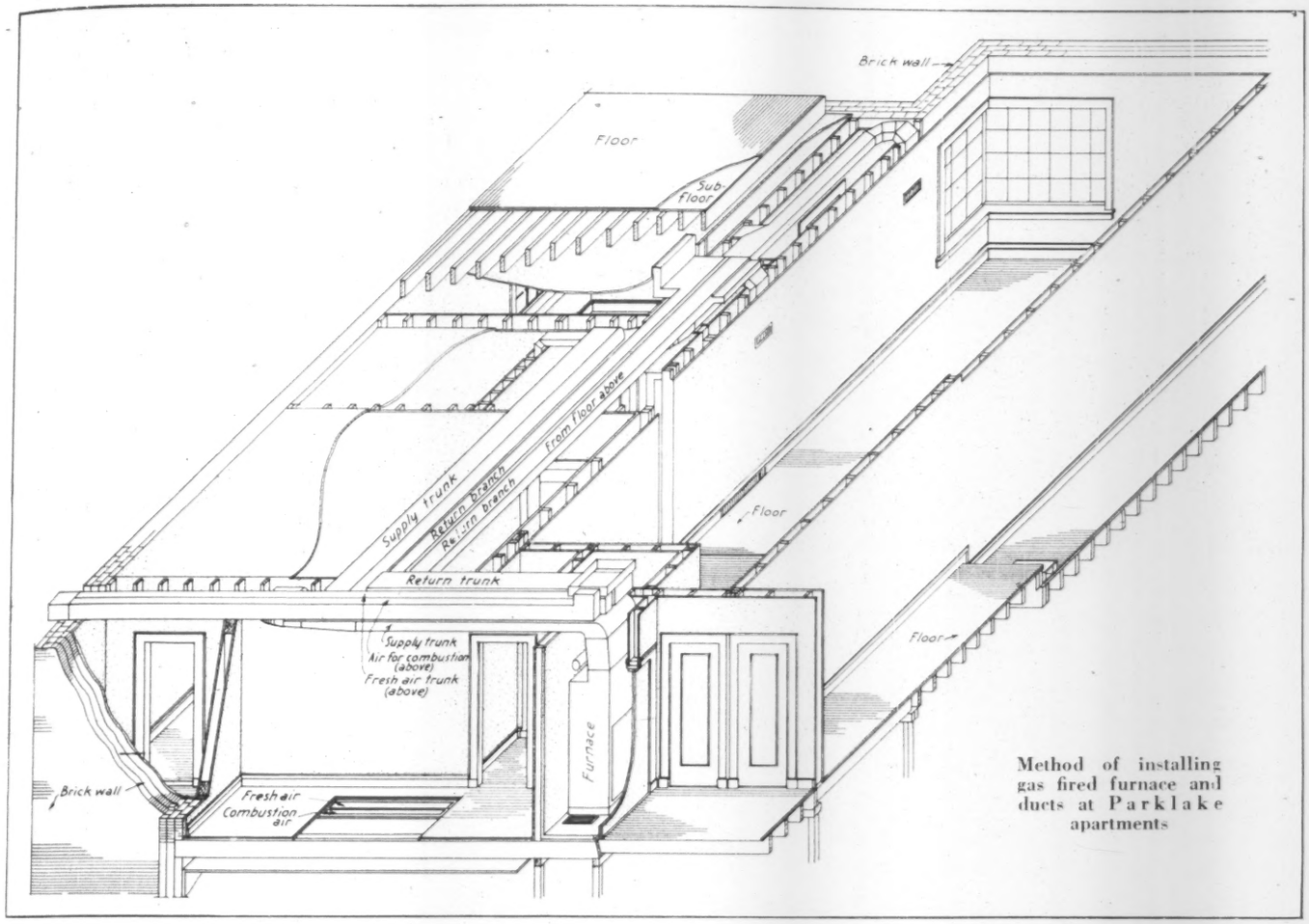
1-2-3. Outside air supply recirculation and exhaust dampers, all mechanically interlocked so as to give 100 per cent outside air and exhaust, or 100 per cent recirculation or any intermediate percentage, under control of master duct thermostat readjusted by outdoor thermostat (arbitrary manual adjustment of proportions is possible)

4. Outside air intake
5. Recirculation duct
6. Outboard exhaust
7. Automatic air filter
8. Steam tempering convector
9. Water chilled convector
10. Humidifier
11. Bypass and face dampers
12. Supply fan
13. Duct reheaters, one for every diffuser
14. Room diffuser and vent grille—about 25 per story for four stories
15. Exhaust-return air ducts
16. Exhaust fan

17-18. Outside thermostat readjusts thermostat 18 which controls air delivery temperature with bypass dampers, 11, and convector valves the year around

19. Individual room thermostat which controls individual duct reheater, 13





Method of installing gas fired furnace and ducts at Parklake apartments

Each Tenant in New Apartment Building Has His Own Gas Fired Heating System

By C. H. Cummings*

THE new Parklake apartment house in Chestnut Hill, Mass. suburb of Boston, is heated and winter air conditioned by a unique method; each individual apartment has its own gas fired furnace controlled by a thermostat in the living room, and supplying filtered outside and recirculated air. Each tenant thus has independent control of his heating system as he would in his own home. Ten per cent outside and 90 per cent recirculated air is furnished during the heating season, and adjustments may be made for taking in larger quantities of outside air during the summer for ventilation. Air intakes are situated in the outside walls of the building, and a duct system carries the air to the heater for each apartment. Total duct work for the building weighs approximately four tons.

The amount of air circulated varies with the size of the apartment, from 380 cfm for the small apartments to 750 cfm for the large ones. Supply grilles are 6 ft, 6 in from the floor and return grilles are in the baseboard. The grilles are of the five direction flow type. The area of the one room apartments is about 500 sq ft, the 2½ room units have 800 sq ft of area, and the 3½ room apartments approximately 1000 sq ft. Total floor area of the entire building is about 137,400 sq ft, and there are 27 suites on each of five floors above ground and 16 suites on the ground floor.

Humidification is by pan type humidifiers and the amount of humidification is controlled only by the water level in the pan.

Corridors, lobbies and stair wells of the building are heated by direct cast iron radiation supplied with steam from gas fired tubular boilers which also supply the

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